HEAT AND HEAT-ENGINES.

A STUDY OF THE PRINCIPLES WHICH UNDER-LIE THE MECHANICAL ENGINEERING OF A POWER PLANT.

BY

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ERRATA IN EDITION OF 1899.

Page 15, line 10 from top, *for* heat unit *read* degree

" 79, heading of 8th column should read:

Calorific Power
observed
in Calories.

" 79, heading of 10th or last column should read:

Calorific Power
calculated
in Calories.

" 86, line 5 from bottom, equation should read:

$$Q = w' \times c' \times (t_1 - t_2)$$

" 99, line 9 from bottom, *for* reducing D to 32° F. *read* reducing D_c to 32° F.

" 104, line 9 from top should read:

$$E = D^2 - \tfrac{8}{12} D + \tfrac{1}{9} = A - \tfrac{8}{3} \sqrt{A} \text{ nearly}$$

" 104, line 11:

$$E = \frac{\pi}{4} \left(D^2 - \frac{8}{12} D \right) = A - 0.592 \sqrt{A}$$

" 107, line 10 from top:

$$V_c = V_a \frac{T_e}{T_a} = 6000 \frac{1085}{521},$$

whence

$$Vc = \mathbf{12,500} \text{ cubic feet}$$

" 107, line 6 from bottom, *for* it BELOW the fire *read* it ABOVE the fire.

" 110, line 7 from bottom:

$$\therefore p = H(D_1 - D_2)$$
$$= 3.51 \text{ lbs. per square foot}$$

and

$$p_1 = 3.51 \times .192$$

" 142, line 11:

$$V = w \times c \times (t_1 - t) = w' \times c' \times (t - t_2)$$

" 142, line 4: grew less intense by *square of* distance

" 165, line 5 from bottom, *for* by multiplying by 778 *read* by dividing by 778

" 180, line 21, equations should read:

$$p_1'v_1 = p_0 v_0 (\mathbf{1} + at_1)$$

and

$$p_2'v_2 = p_0 v_0 (\mathbf{1} + at_2)$$

" 181, lines 14 and 15, *for* inversely *read* directly.

" 207, line 1, column 1, top of page, *for* 24.3 *read* 25.7

" 216, line 3 from bottom:

$$\theta = \int \frac{dq}{T} = \text{etc.}$$

" 224, line 6 from bottom should read: If, then, from the preceding paragraph, the total heat re-

" 226, line 12 from bottom should read: an evaporation of 34.488 pounds

" 229, line 7 from bottom:

$$PLA = \frac{P \times 144 A \times L}{144} = paL$$

" 236, line 3 from bottom: the mean pressure p_m results

[*Over.*]

Page 236, last line:

$$\frac{A}{7} = p_m = \text{etc.}$$

"   241, line 3:

$$W_1 = p_1 v_1^{n} \int \text{etc.}$$

"   243, 4th equation from top:

$$\left(\frac{V_1}{V_2}\right)^n = \text{etc.}$$

"   243, 6th equation:

$$\frac{p_2 v_2}{p_1 v_1} = \text{etc.}$$

"   244, lowest equation:

$$E_1 - E_2 = W_1 = Jc(T_1 - T_2) \times l$$

"   256, line 20:

$$144) \div 0.3148, \text{ the latter, etc.}$$

"   261, 4th equation should read:

$$= p_1 v_1^n \int_{v_2}^{v_1} \frac{dv}{v^n} = \frac{p_2 v_2 - p_1 v_1}{n - 1}$$

"   263, first equation of Art. 183 should read:

$$n = \frac{C_p}{C_v}$$

"   290, equation middle of page should read:

$$le : lk \; \therefore \; V : V_1 \quad \text{or} \quad le = \frac{V}{V_1} lk$$

"   301, equation middle of page should read:

$$H_3 = C_v T_2 \text{ hyp. log } r.$$

"   301, below this:

$$H_4 = C_v T_1 \text{ hyp. log } r.$$

"   306, line 17 from top: the ϕ is not a subscript, but should be same size as T, and should read $T\phi$

"   308, line 12, *for* Water per H.P. per hour *read* Work per H.P. per hour

"   331, line 3 of Art. 233, *for* values *read* valves

"   332, bottom line should read:

$$LK = NA = f(AL), \quad \text{since} \quad \frac{LK}{KI} = \frac{C}{AL} \quad \text{and} \quad \frac{NA}{KI} = f$$

"   333, third equation from top should read:

$$\frac{1}{r} = \frac{NB}{LI} = \frac{AB + NA}{LK + KI} = \frac{AB + KIf}{KI + KIf} = \frac{\frac{1}{r} + f}{1 + f}$$

"   351, second equation from bottom:

$$Q_e = Q + Q_j - Mq_4 - G(q_k - q_i) - AW$$

"   351, last equation should read:

$$Q_e = H_2 - H_3 - Q - Q_j + Q_e + A(W + W_c)$$

"   368, lines 12 and 13 should read: perature T_4 in this last equation. That is, place

$$W_4' = 778M(C_p - C_r)T_3\left(1 - \frac{p_4}{p_3}\right) = 778MC_p(T_3 - T_x)$$

"   372, bottom, and page 373, top: in these equations what appears as Cp should *read* C_p

"   391, line 1 of (3), *for* compresses *read* displaces

"   401, line 4 of Art. 274, *for* stant temperatures *read* stant volume

PREFACE.

THE effort has been made under another title, "The Mechanical Engineering of Power Plants," to discuss the steam-engine and the steam-boiler with their accessory apparatus, so as to enable students and others to make an intelligent selection of successful designs for a set of conditions which might be imposed. The object of the author was to make the reader familiar with accepted solutions of the power-house problem in all its details, but no attempt was made to discuss the questions of design of such apparatus. It was intended that the student should ask at the end of his study of that book: What are the principles of physics and dynamics upon which these machines depend; and how do engineers proceed when called upon to design such power-house engines ?

This book, under the title of "Heat and Heat-engines," has been prepared to answer these questions in part. It discusses the energy resident in fuels, and the methods of its liberation as heat for power purposes; the transfer of such heat to convenient media whereby it can be used in heat-engines; the laws and properties of such media, and the design of cylinders of the necessary volume to give a desired mechanical effect or horse-power. Then, this point having been reached, and relations being established for the mutual variations of temperature with pressure and volume in such media when operated in a cylinder with a piston, it becomes

easy and natural to go farther and discuss the air-compressor and its complement, the air-engine; and to extend this discussion to include the problem of mechanical refrigeration. The hot-air engine using a permanent gas as a medium naturally leads to the gas-engine and the oil-engine; and the engine using steam as a medium leads naturally to those using other media, such as naphtha, alcohol, and ammonia. The chapter on the Injector as a heat-absorbing and energy-transforming device closes the book.

If the distinction were a conceded one, the first book designated as the *mechanical* engineering of power plants might be said to lead to this supplementary discussion upon the *dynamical* engineering of power plants. The treatment assumes and does not attempt to prove the dynamic or mechanical theory of heat, and does not ask nor require to know whether there is an intermolecular ether or not, nor whether the energy of heat manifests itself by producing a vibratory or undulatory or vortex motion of matter, or is an electro-magnetic phenomenon. These discussions belong to a transcendental sphere of investigation and research with which the practical engineer as a rule need not concern himself.

It is somewhat in this latter view also, that for the purpose in hand the term "thermodynamics" has been largely avoided, as well as the attractive development of the truths of the science of heat-engine design by the methods of exponential equations and the use of the calculus. No one is more ready than the writer to recognize the elegance of the deductive method from fundamental equations, and the delights of the revelations of law which are thus secured. But on the other hand it must not be overlooked that the very ease and elegance of the deductive method makes it an unsafe tool in the hands of the inexperienced who are without the steadying effect of long familiarity with the actual conditions of the applications of theory, which should prevent the drawing of conclusions which the mathematical treatment

alone does not pronounce unsound. The science of thermo-dynamics has this singular distinction, that it has been developed deductively from fundamental equations by master-minds, using the methods familiar to the physicist and the skilled mathematician. But it has often moved so far in advance of the ordinary attainments of the practitioner in the power-house on the one hand, and so far in advance of the experience of the student on the other hand, that both have often lost the connecting-link between the advanced theory and the current practice. The effect on the student has been to weaken his appreciation of the subject if he was not of a mathematical bent; or if he was gifted with facility in this method of thought, he failed to satisfy his early employers because he applied the deductive methods and conclusions with a zeal not always according to knowledge. The practitioner, on the other hand, as the result of years of training in the inductive method of generalizing from particulars in his daily affairs, is out of touch with the deductive method and has no use for the unfamiliar process and its practitioner. He therefore unjustly undervalues his young technical graduate and his method of training.

This treatise tries to occupy a middle ground. It might wisely be used as a groundwork for a subsequent treatment of heat-phenomena by the analytic or mathematical method after the student has become familiar with the physical facts of which the equations of thermodynamics are condensed statements. By pursuing this middle course, however, a few places may be detected where the logical mind will miss the antecedent premise upon which the conclusion is based, or where it is stated upon authority, and the proof is not given. This is the result of trying to treat thermodynamics without the calculus, and the result should be to turn the student to further and exhaustive research in the higher field. At least, this is the author's desire.

Equations could not be avoided, nor the use of loga-

rithms; but the use of the temperature-entropy diagram for the graphical representation of relations has been abundantly permitted. The appendices also open doors into departments of further research and knowledge beyond the scope set by the limits of the text.

The student and writer of to-day is the heir of the work and thought of his predecessors who have inspired and guided him. The bibliography in the appendix will show how many workers in these lines have left their impress upon the modern treatment of the subject. The influences most felt in this book are those from Rankine, Cotterill, Ewing, Peabody, Wood, Reeve, and Richmond, to whom heartfelt acknowledgment is extended and to whose treatises the advanced student is referred. To the last name in the list special thanks are due for valuable suggestions and a criticism upon certain parts in the proofs. To Profs. Thurston, Carpenter, and other contributors to the Transactions of the American Society of Mechanical Engineers, and to Prof. Reeve, the author is indebted for use of helpful illustrations.

The specialist will require to pursue the lines of his selection by supplementing the basal treatment of this book by further study in the excellent treatises on the gas- and oil-engine, the injector, the refrigerating-machine, and in the field of the application of compressed air. It will be a great pleasure if the treatment and its methods shall make the book useful also to that growing class of persons who are brought into touch with engineering matters and are anxious to learn about them, and yet who are not fitted to profit by an exclusively mathematical discussion. The object which has been sought in its preparation will be secured if the book shall prove helpful and useful as a stimulus to further study.

Columbia University, New York, *July*, 1899.

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CHAPTER I.

INTRODUCTORY.

1. Sources of Motor Energy.—There are three great sources of force or energy for industrial uses. The first to be utilized is the force resident in the contractile tissue of the muscles in man and animals, which is known as muscular force. The second is called the force of gravity and is the force by which the earth attracts all masses toward its centre. The third is the group of forces which become manifest or are released upon chemical reactions such as combustion or oxidation; the two most important of these latter are the forces of heat and electricity.

2. Limitations of Muscular Force and the Force of Gravity.—While the muscular force of men and animals varies with the race, species, size, health, training, temperament, and muscular endowment of the individual, yet certain fixed limits are set to the amount of energy to be gotten from any single unit. Large powers can only be obtained by aggregating many units, which is inconvenient and costly; but more than all, a limit is set by the endurance of the animal unit, which must have periods of rest and recuperation. Speed is also limited by the ability of the animal motor to

maintain a maximum effort for any length of time.   Finally, there is no considerable reserve store of energy to be drawn upon if more effort is required.

The force of gravity becomes available as a motor force when a weight or mass is lifted to a higher level and is permitted to descend to a lower one.   Solid weights are only serviceable when lifted by some other mechanical force; water and air are the only weights which are otherwise lifted further from the centre of the earth independent of man.   The former is lifted by the sun in vapor to high levels of land, whence it seeks to descend to tide-water level again; and the winds are produced when colder and heavier air descends and displaces the lighter air which the earth has warmed. Gravity, therefore, as a motor force is dependent upon the availability of higher levels of land at which a sufficient mass of water can be accumulated, and an adequate reservoir in any particular region or an adequate flow from a source is a necessary condition for the use of water-motors; and while there is an abundance of energy present in the atmospheric ocean at the bottom of which all industry is carried on, yet at present the reliability, controllability, and capacity which must belong to the satisfactory working of an industrial motor are lacking to windmills in most places except where used for pumping or where they can be used to store some other form of energy in accumulators.

This same series of difficulties has beset the successful application of the energy stored by the winds and other disturbances in the ocean waves.   Tide-motors depend upon the lifting of the ocean level by the stellar or planetary attractions, and are reliable and controllable, although only made of large capacity at great cost; but the types of motors as yet devised to use the impact or lifting force of coast-waves have not proved reliable or permanent enough for engineers to venture to adopt or install them.

Since it is the sun's heat energy which lifts the water

and disturbs the equilibrium of the strata of air, it would appear that water-motors and windmills are in this sense heat-motors in the last reduction.

3. Importance of the Motor Energy Liberated on Combustion.—It will be at once apparent, on a moment's thought, that while the energy resident in falling water is most serviceable and is destined to become more so as the means of transmitting energy are improved, yet there are many causes which have acted to make the use of the steam-engine, the gas-engine, and the hot-air engine by far the most widespread at this time. The energy in falling water with a few notable exceptions is limited in amount both by the weight available and by the height of fall; while in combustible fuel or oil or gas there is stored an amount of available energy which is practically unlimited where the supply of combustible is not limited. The energy, moreover, is in a compact bulk; fuel can be had native in many regions where there is no head of water, and where fuel is not native it can easily be transported. It will be seen, therefore, that the study of those forms of motors which are so widespread compels the study of the laws and principles which underlie the phenomena of heat, and that the general name of heat-engines may properly be applied to such engines.

While every one believes that the near future is to reveal methods for generating or liberating energy directly from fuel in the form of electromotive force, and this is now done by the chemical reactions which occur in various electric batteries, yet at this writing the importance and extent of the applications of such methods make them lie in the province of the physicist still, rather than in that of the engineer.

4. Analysis of a Power Plant.—The industrial result in a power plant is the production of something which shall have a commercial or salable value. This may be a manufactured article, or it may be a safe transportation of goods or persons for which the community shall be willing to pay. Hence the

last link in the power-plant chain will be as extensive as the entire field of industry.

The first step or link is the generation or liberation of energy to furnish the necessary power. In the heat-engine this occurs where the combustible fuel is burned in a proper furnace or fire-box. This liberated or generated energy must be suitably stored in a reservoir or accumulator from which it may be drawn off as required. In a steam plant this function is discharged by the boiler; in gas-engines this storage step is lacking. In water-power plants the liberation and storage are done for the engineer either before his work begins, or else the storage reservoirs or dams become very important features of his undertaking. The third step or link is the appliance whereby the energy stored in the second step, and held as potential energy, shall be transformed into actual energy by being permitted to act through a prescribed path under the control of a capable intelligence. This appliance is the engine or motor, which must be adapted to the force which is to be utilized both as to capacity and as to form, and which demands a knowledge of the laws and properties which attach to the medium whereby the energy is revealing itself. Fourth in the chain comes the machinery of transmission, whereby the motion of the motor or engine and its developing power shall be adapted or properly transformed to meet the uses and purposes of the machine whereby manufacturing or transportation is effected at the industrial end of the series.

The subject of transmission of power is by itself so important and extensive, and the industrial field so limitless, that power-plant study may properly be limited to the other or first three steps for convenience. The water-motor and the wind-motor will also be excluded from present consideration for the sake of confining the scope of study. The field will therefore become that which embraces the generation or liberation of energy in the form of heat, and the utilization of that energy in the heat-engine.

5. Scheme of Classification.—In this view the subject of heat and heat-engines will group itself for study under the following heads:

   I. General Notions on the Phenomena manifested in Heat-engines.
  II. Sources of Heat, or Fuels; Generation or Liberation of Heat. Combustion.
 III. Transfer of Heat. Heating-surface.
  IV. Media used to Transfer Heat Energy to Engine-orgars. Properties and Physical Laws.
   V. Work done by a Heat Medium; Relations of Heat and Force in Expansion and Compression. Cylinder Design.
  VI. Thermal Analysis of a Heat-engine. Heat Cycle and Efficiency.

———

 VII. Air-Compressor and Compressed-air Engine.
VIII. Hot-air Engines.
  IX. Internal-combustion Engine. Gas- and Oil-engines.
   X. Vapor-engine.
  XI. Mechanical Refrigeration.
   A final chapter on the Injector is appended.

It will be observed that the latter sections are in a sense to be viewed as the fuller application of the principles discussed in the first six headings.

CHAPTER II.

GENERAL NOTIONS ON THE PHENOMENA MANIFESTED IN HEAT-ENGINES.

6. General and Introductory.—In order that the engineer may have a satisfactory form of motor, it must be one in which an adequate force acts through the desired space in a unit of time. A force of illimitable extent is of no practicable value unless it is exerted through a finite and measurable space. Hence every real engine has an organ capable of receiving the action of a force or effort, and capable of moving through a constrained path under the action of that effort, while the force is overcoming the resistance moving through the required distance. In countries which use and prefer the metric system, the unit of force is the kilogram, and the unit of path is the meter. The product of effort into its path is called *work*, and the unit of work is a kilogrammeter. Where the pound is the unit of force, and the foot is the unit of path traversed by the effort, then the work will be expressed by their product as before, but the work-unit will be in footpounds. For a large output of work from a motor, the footpound or kilogrammeter is inconveniently small; hence multiples are usual. The accepted standard as evaluated by James Watt from experiment is that called a " horse-power," and is equivalent to a work of 33,000 foot-pounds done in one minute. The equivalents of the horse-power are:

6

Horse-power.	English Foot-pounds per Minute.	French Kilogrammeters per Minute.	Austrian Foot-pounds per Minute.
English and American....	33,000	4,562.46	25,233.6
French	32,548.2	4,500	25,420.8
Austrian	33,034.2	4,567.14	25,800

The metric horse-power is more usually expressed as 75 kilogrammeters per second. The unit of electric output is known as the Watt, and represents a work in foot-pounds equivalent to $\frac{1}{746}$ of a horse-power. One thousand watts or a kilowatt is therefore equal to 1.34 horse-power.

In heat-engines the universal practice is to have the effort which overcomes the resistance take the form of a pressure of a gas or vapor exerted upon a given area. It will appear hereafter how and why the heat produces pressure. If the elastic tension of the gas or vapor be expressed in pounds per square inch, and its pressure is exerted upon a disk or piston which fits a cylinder without leakage and which has an area expressible in square inches, then the product of the pressure into the area in these units will give a total effort in pounds. If the unit be the kilogram per square centimeter, and the area be in square centimeters, the effort will be in kilograms. The piston therefore must move through the necessary number of feet or meters per minute, in order that the horse-power required may be generated. Hence if S denote the space in the desired units through which the effort PA in the corresponding units is made to move in one minute, then the total work of one minute will be PAS in the required compound unit of foot-pounds or kilogrammeters. Since the usual heat-engine cylinder is of comparatively short length for convenience of construction, the common requirement is that the piston which fits it must traverse its short length many times in one minute. If L denote the length of the traverse of the piston in meters or feet, and N denote the number of times this traverse is made in one minute, then the

initial S above may be replaced by LN; whence the work-expression becomes

$$\text{Work} = PALN$$

in whatever units are used. Since the horse-power of the engine will be as many as the number of times that the unit 33,000 is contained in the total work in foot-pounds, the above will become

$$\frac{\text{Work}}{33000} = \text{H.P.} = \frac{PALN}{33000}.$$

Furthermore, if A be expressed in square feet instead of square inches, then AL represents the volume of the cylinder in cubic feet effectively filled at the end of each traverse of the piston, and if N denotes the number of times per minute that this volume is filled, it follows that the work-expression can be written

$$\text{Work} = PLAN = PV,$$

when V denotes the volume filled in a unit of time by the working vapor or gas which causes the pressure P in the cylinder. The latter now must be given in pounds per square foot. PV is therefore a very general expression for work in foot-pounds per unit of time. It can also be used to express the foot-pounds of work done in one stroke of an actual engine if N be called unity, or the work done by one unit weight of vapor which occupies a specific volume V.

A steam-, gas-, air-, or vapor-engine operating so as to make the foregoing discussion apply to it may conveniently be designated by the general name of a *piston-engine*, and this term will be generally used for such machines in the discussions which follow. It applies equally well to a pump in which a fluid resistance represented by P in pounds per square foot is overcome, and a volume V of the fluid in cubic feet is displaced in one minute, whether by piston or by plunger.

7. Graphic Representation of the Work of a Piston-engine.—Since the work of a piston-engine is the product of

two factors—feet multiplied by pounds, or foot-pounds,—it is obvious that a closed figure can be drawn enclosing an area which, upon an assumed scale of units, shall be the same as the given product in foot-pounds. Furthermore, whatever the shape of that figure, a rectangle can be drawn the product of whose base into its altitude will give the same area, or represent that same number of foot-pounds of work.

In piston-engines it will become manifest later that the figure representing the work takes a small number of typical shapes. It is convenient to agree to represent pounds by the vertical ordinates, parallel to the coordinate axis of Y, and to represent feet by the horizontal abscissæ, parallel to the coordinate axis of X. A piston-engine can then be made to draw its own work-diagram by a simple device. If the pressure P from the storage source of supply which is to force the piston forward be also let into a small cylinder of known area of cross-section, and bear upon a piston in that cylinder whose motion is resisted by a calibrated spring, then the piston effort and spring distortion will balance at a certain point. If a marking-point or pencil be attached to the spring piston, the position of equilibrium of pressure and spring can be marked and noted. Further, if the motion of the engine-piston be given to a board or drum in a horizontal direction, while the pencil which is controlled by the calibrated spring travels vertically, the condition is fulfilled of having the horizontal dimensions of the traced diagram represent feet or be proportional to feet, while the vertical dimensions represent pounds or are proportional to pounds. An instrument embodying this principle, and modified to conform to conditions of convenience and accuracy, is called the Indicator.

If the P.V. form of the expression for work is preferred, then the horizontal lines will be proportional to volumes. The linear length of the engine-stroke must be multiplied by a factor representing the area A of the engine-piston. In the first case, the area of the work-diagram gives the work per each square inch of area of the engine-piston, and must

be multiplied by the same area A to give the work of the entire piston.   Hence either method may be followed as is most convenient.

The simplest case of work-diagram is presented in Fig. 1.
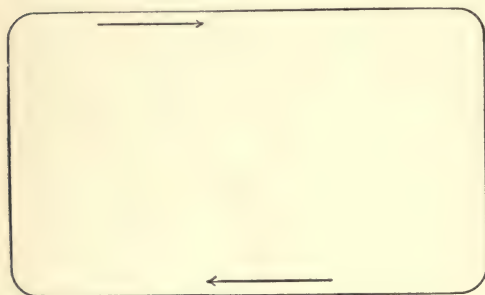
FIG. 1.

Here the pressure P from the boiler or reservoir is constant throughout the stroke of the engine, and the diagram is essentially a rectangle. If, however, the pressure is not constant (and it will be seen hereafter to be desirable that it should not be) throughout the whole stroke, then the general form of the diagram will be that of Fig. 2.   Here, beginning at
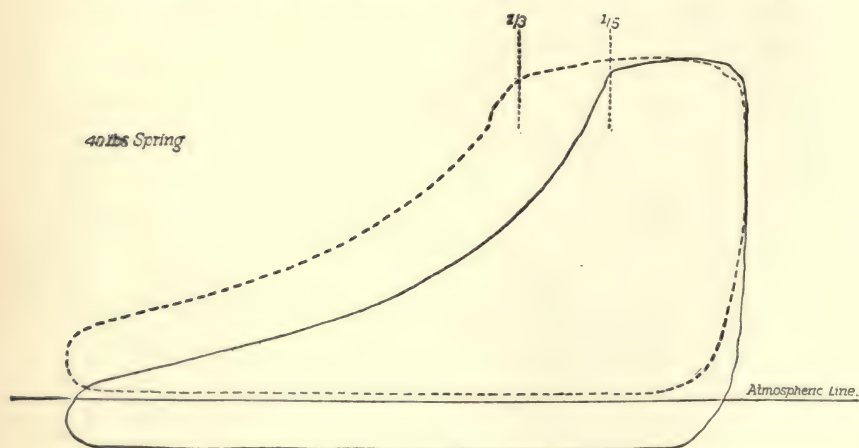
FIG. 2.

the upper right-hand corner, there is an admission at constant pressure up to the end of the upper horizontal line, and

then a fall of pressure gradually, as given by the curved lines, indicating a variable or varying pressure down to the lower left-hand end. The curves at the right and left sides of Fig. 2 indicate a variation of pressure with volume of cylinder; and the law of such variation (if there is one) has obviously a considerable effect on the work done in a piston-engine per stroke. It further deserves study to ascertain what it is which causes a variation in the vertical distance apart of the upper and lower lines of the work-diagram and what are the laws of its action. These obviously affect also the power to be gotten from the engine.

Here, then, is the problem. The proportions of the work-diagram in a heat-engine as to size and shape are affected by heat. What is heat, and what are its laws and principles?

There are other diagrams which may be drawn to present the operation of heat-engines, besides the pressure-volume diagram above discussed. These will be referred to in their proper connection hereafter, and after the indicator (or P.V.) diagram has been considered.

CHAPTER III.

GENERAL NOTIONS ON HEAT.

8. Introductory.—It should be stated at the very outset of study that the essence or nature of what is called heat is not known. Like the nature of the force of gravity or the vital force, a mystery as yet impenetrable shrouds everything concerning it except the phenomena which it occasions. Hence it has been the function of the physicist and mathematician to find hypotheses or assumptions which shall explain and agree with the phenomena, and which shall further enable predictions to be made as to the results to be anticipated with untried combinations. Such hypotheses or theories must explain and agree with *all* phenomena if they are to be acceptable, and one discordant or unexplained phenomenon is sufficient to throw doubt on the working theory then in general acceptance.

It belongs to the province of the physicist rather than to that of the engineer to review the theories concerning heat which have heretofore prevailed. Fortunately it is not necessary for the engineer to be conversant with the refinements of theory to be able to use its general principles with intelligence.

The universally accepted theory of heat is based upon the postulate which is known as the Conservation of Energy. This announces that force is as indestructible as matter, and that the most which happens to energy when apparently dissipated is its conversion into other forms of energy, just as

matter is converted into other forms of matter when a combination is apparently destroyed. This theory is known as the Dynamic Theory of Heat or the Mechanical Theory of Heat, and its axiom is that "Heat is a Mode of Motion."

9. Mechanical Theory of Heat.—The phenomena of the conversion of work into heat have been long observed in the effects of impact, of heat from rubbing friction and abrasion, and the like. Rumford's historic experiment (1798), Davy's investigations (1799), and the work of other physicists led up to the work of Dr. Julius Robert Mayer of Heilbronn (1832–1842) in Germany, and the quantitative research into the convertibility of work into heat by Dr. Joule of Manchester, England (1843). Later Sir William Thomson (1850) extended analogous principles to electricity, and the name Thermodynamics has been applied to the science which is concerned with the relations between heat and mechanical energy under all conditions.

The fundamental law of thermodynamics, sometimes called the first law, is that *Heat and mechanical energy are mutually convertible; and heat requires for its production and produces by its disappearance mechanical energy in the proportion of 778 foot-pounds for each British thermal unit.* This law is physical and experimental; it is a deduction from phenomena and tests, rather than an intuition or an axiom.

10. The Mechanical Equivalent of Heat.—The factor 778 was originally determined by Joule to be 772; later determinations give it the higher value (Rowland). Joule's experiment was to find the weight which, falling through one foot in height, would produce an amount of heat sufficient to raise one pound of water one degree on the Fahrenheit thermometer-scale. This factor is called the mechanical equivalent of heat; in metric units it has a similar definition, but the value of the factor is 426.8 kilogrammeters per kilogram of water raised $1°$ centigrade. Out of respect to its first investigator it is usually designated by the first letter of his

name, and is represented in formulæ by the initial J. Its reciprocal, or $\frac{1}{J}$, is often designated by the capital letter A.

11. The British Thermal Unit.—In enunciating the first law of the accepted theory of heat, the words British Thermal Unit were used. What is a thermal unit?

The thermal unit, or unit of heat, is the quantity of heat or the corresponding energy in foot-pounds which will raise a unit of weight of water through one degree of the accepted scale of a thermometer. The British thermal unit is therefore the quantity of heat or energy which raises one pound of water one degree Fahrenheit; the metric unit is the quantity of heat or energy which will raise one kilogram of water one degree on the centigrade scale. The metric unit is called the calorie and is 3.968 (roughly 4) times the B.T.U. Strictly, the B.T.U. is the heat required to raise one pound of water from 39° to 40° Fahrenheit, and the calorie is the heat necessary to raise one kilogram of water from 4° to 5° centigrade. These figures are those at which water has its greatest density from experiment. Many engineers and writers use the temperature of melting ice as the starting-point, and recent British and French authorities prefer to use 62° Fahrenheit or 15° Centigrade as the base. This difference will explain some discrepancies among accepted authorities upon these questions.

12. Specific Heat.—It must follow from § 9 that if heat and energy are mutually convertible, then different bodies must vary with respect to their capacity for receiving, storing, and giving out this energy. In the general field of mechanical science it has been found that the measure of stored energy in a moving organ of a machine or a free body is made up of the product of its Mass by the half-square of its velocity of motion $\left(\frac{Mv^2}{2}\right)$. In molecular or atomic motions such as those in question in heat-motion, the same conceptions

are supposed to apply, the only differences being the infinitesimal character of the atomic mass, and the probably inconceivably great velocity of the motion—whatever it may be. Hence the mind is ready to accept the observed fact of such great differences in the thermal capacities of different bodies, and also the differences in the same body in different states.

Speaking generally, then, the quantity of heat or energy which is required to raise a unit mass of a substances by one heat-unit will be called its specific heat. It will be seen hereafter that this general statement needs to be guarded in cases where any other change occurs by heat applied to a body besides an increase in its temperature (see § 116). The specific heat of bodies, solids, liquids, vapors, and gases requires to be experimentally determined by the physicist in the laboratory. Tables of specific heats appear in an Appendix. Water has a specific heat of unity by agreement among experimenters, both because its value is so large, and also because it is so conveniently used in comparisons and transfers.

13. **Temperature.**—It follows from the mechanical theory of heat and the above discussion that temperature, as measured by the appliance called a thermometer or by the human sensorium, is not a measure of the amount of heat energy resident in any two or more different bodies or masses. It is an indication of its intensity, however, and of that portion of the total energy which can affect the senses of the observer. An experiment to show this is made by taking equal weights of two bodies like iron and water at the same temperature, and putting them into another quantity of some liquid at another temperature. The water will transfer much the more heat to the liquid.

It will appear shortly, however, that the product of the mass or weight by its specific heat and by its temperature (the latter being properly observed) will give a measure of the

heat energy. Of two bodies of the same substance and therefore having the same specific heat, but being at different temperatures, and near each other, the hotter body tends to become cooler, and the cooler body becomes hotter by the receipt of transferred heat-motion.   Heat passes of itself from a hotter to a cooler body, but this process does not reverse except by introducing a factor of mechanical energy to cause it to do so.

14. **Thermometers.**—Appliances for measuring or observing differences in temperature (but not differences of heat necessarily) are called thermometers.   Most of them depend upon the property of a liquid—mercury or alcohol—whereby it expands equally for constant increments of temperature (see § 113).   If the liquid is confined in a tube of fine calibre, the expansion is easily read on a properly graduated scale. Solids have this same property of expansion by heat, and can be used for higher temperatures.   They are then often called pyrometers.   Gases can also be used in thermometry.

Without entering too deeply into this subject, there are two fixed points of temperature which are used in graduating thermometers: the point at which ice melts, and the point at which water boils under a pressure of one atmosphere.   The latter is that given by Regnault's determinations, of 14.7 pounds per square inch above vacuum, or 2116.2 pounds per square foot.

The Fahrenheit scale calls melting-ice temperature that denoted by 32° on its scale, and boiling-water temperature is 212°.   Its zero-point is thus 32 degrees lower than melting ice.

The centigrade scale divides the 180 degrees Fahrenheit between melting ice and boiling water into 100 parts, and places its zero at the melting-ice point.

The Réaumur scale divides the 100 degrees of the centigrade scale into 80 parts.

The transformations from one scale to the other are not difficult.   See Appendix.

15. Air-thermometer.—At high temperatures the liquids used in thermometer-tubes become vapors, and at low temperatures the liquids freeze. This change of state of the registering body is not only inconvenient, but is accompanied by inaccuracy and uncertainty near both limits. Hence it has been sought to use a permanent gas as a thermometer, and air has been most used by reason of its convenience and adaptability.

In principle the air thermometer consists of a perfectly cylindrical tube, closed at the bottom, and containing a quantity of air below a bubble or drop of mercury in the tube, which is to serve as a register for the expansion of the air below it. If the air-volume below the mercury be exposed at the pressure of one atmosphere (the barometer reading 29.922 inches) to the temperature of melting ice, and the position of the bottom of the mercury telltale is marked, and then to the temperature of boiling water, and a similar mark made there, the range for the 100 or 180 degrees of the usual thermometer is given, and the fixed points are determined. Regnault's investigations showed that for a length of air-column below the lower mark, represented by unity, the length from the bottom to the upper mark would be 1.3665—that is, the expansion between these limits is 0.3665 of the original volume. Hence it would appear possible to graduate such a tube by means of this property, assuming that no changes in other properties occur at wide ranges beyond the range of experience. The graduation upward for high temperatures offers nothing unusual, but the graduation below the point of melting ice leads to an interesting inference.

If the Fahrenheit degrees be used for the length of 0.3665 times the unit length, then each degree of the air-thermometer has a length

$$\frac{0.3665}{180} = 0.00203611$$

FIG. 3.

Air Thermometer: constant volume: Safe up to 800° Fr.=say 425° C. Scale, ½ full size.
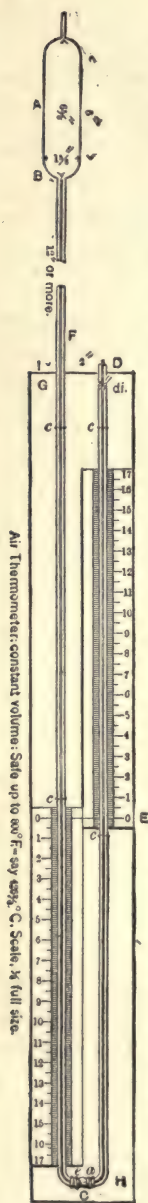
of the unit length below the line marked for melting ice, or for each degree Fahrenheit of increase in temperature or decrease the air-volume increases or decreases by that fraction of its length. If it decreases by $\frac{1}{.00203611}$ of its length for each degree Fahrenheit, then at a temperature Fahrenheit represented by 491.13 below 32°, or 459.13 below zero, a temperature must be reached at which the property of further reduction of volume by withdrawing heat or heat energy disappears. In other words, the body appears to have no heat energy at that point. This is called, therefore, the absolute zero of the air-thermometer. If the degrees be numbered from this point, ice melts at 491.13° and boils at 671.13°. For centigrade scale of degrees the air-thermometer zero is 273° below the zero of the ordinary scale at melting-ice temperature. Fig. 3 illustrates a form of air-thermometer, the result of effort by Mr. Fred W Prentiss and the late J. C. Hoadley.

16. Absolute Temperature.—If air were a perfect gas, or one which would expand exactly so that its rate of expansion would be the same as that at which it absorbs heat, the air-thermometer scale could be used as an absolute temperature scale. It is practically so, and will be so used hereafter. The only error comes in determining the zero of the absolute scale. The work of Joule and Thomson (1854) and that of Rowland (1879) have shown the discrepancies between the real and ideal readings (see Appendix).

The absolute zero as computed for a perfect gas is 492.66° F., or 273.7° C., below the melting-point of ice.

The significance and usefulness of the absolute scale is very great in work with heat and heat-engines. In fact it is indispensable. From what has preceded (§§ 9, 12, 13) the total heat energy present in any body at any time will be the product of its weight by its specific heat by its absolute temperature, when no energy or heat is in process of absorption in doing work upon the body itself. If the mass is a unit of weight, and two differing states of the same body are compared, then the difference in absolute temperature measures the heat energy which has been given out or absorbed under the same limitations.

17. Total and Intrinsic Energy.—It will appear finally from the foregoing discussion that the total energy resident in a body is not always to be evaluated by its temperature or by the thermometer alone. Part of it may be measured when changes of such energy occur, when the specific heat is known, and the initial and final temperatures absolute or ordinary. But such parts of the heat energy as are taken up in changing the molecular motion of the atoms of the body are said to become latent or hidden because the usual appliances do not record them, and by certain other parts of an applied heat energy an external work in foot-pounds may be done which the thermometer will not reveal (§ 111). It is obvious, therefore, that some attention must be directed to the effects of heat energy upon the substances upon which it acts, and the means of producing that energy or liberating it if stored.

The latter will be taken up first.

Intrinsic energy in a body is its capacity for performing work by virtue of the heat energy resident in it, without addition of such energy from without. The zero of heat energy is at the absolute zero. The inner or intrinsic energy at any other absolute temperature will be the product of its weight into its specific heat into the range of its absolute temperature above absolute zero.

CHAPTER IV.

GENERATION OR LIBERATION OF HEAT.

COMBUSTION.

18. Introductory.—To supply the energy in foot-pounds required to overcome a considerable resistance, and to do this for a heat-engine in which 778 foot-pounds shall correspond to one heat-unit, requires that there shall be continually introduced into the heat-engine through a proper organ the necessary supply of heat-units in each unit of time. From what source or sources shall this heat and energy be drawn? While heat appears as a transformation of mechanical energy in friction, impact, abrasion, attrition, and in overcoming electrical resistances, these sources are excluded when the object sought is heat which may itself be transformed into mechanical energy.

19. Heat from Combustion.—The most widespread, convenient, and cheap source of heat has been found as the result of causing the oxygen of the atmosphere to combine chemically at a sufficiently rapid rate with certain other of the chemical elements. Oxygen combines with many of the metals or bases or elements as with iron, manganese, boron, phosphorus, and the like, but these are either too costly to serve as convenient sources for heat, or else the process of oxidation is so slow that sufficient heat cannot be derived from them in a short time.

Combustion may be defined as a combination with oxygen which takes place with sufficient rapidity to be accompanied by the phenomena of heat and light. The elements which are found to possess the affinity for oxygen which is required

for combustion are Carbon and Hydrogen, as elements or as compounds. Sulphur often enters into compounds of carbon and hydrogen, but has an unimportant position, so as to be usually negligible. A carbon or a hydrogen element or a hydrocarbon compound, found native or manufactured so as to be obtainable in sufficient quantities and at a low cost to be used as a source of heat, will be called a *fuel*.

20. Certain Phenomena of Combustion.—It is desirable to secure a greater exactness of conception concerning certain facts in connection with the generation of heat from fuel.

In order that practical combustion of a fuel may occur, it must be '' set fire to.'' *Ignition* is the beginning of active chemical action, and the first step in a combustion. Most combustibles require to be raised at their surface to a certain temperature before this ignition, or '' taking fire '' will occur, or if kept cooled below this temperature of ignition the combination of oxygen will not occur, or will cease if it has before been in progress. This temperature of ignition is quite high for many fuels, and its maintenance has an important bearing upon smoke-prevention. Ignition is most easily secured by a flame, and a flame is one of the indications of such ignition.

A *flame* is a body or current of gas carrying in it solid particles at such a temperature as to glow or give out heat and light. These solid particles are usually carbon in a finely divided state, and the heat of the flame is roughly measured by the degree of the incandescence or glowing of these particles. A red flame is not so hot as a yellow flame, and a white flame is the hottest of all. Flame is produced when the supply of oxygen at the place where ignition occurs is not quite sufficient to form at once a gaseous product of the combustion. When the supply of oxygen is copious and means are taken to heat it and mix it thoroughly with the combustible matter, then the heat is very great at the

point of combustion, but there is little or no flame. The gases are blue or colorless and have little heating power except by contact. The heat from the particles glowing in the flame is given off from a large extent of surface, and for many purposes the flame is preferred to the non-luminous gas-current of theoretically perfect combustion. Purely gaseous substances do not as a rule become luminous by heat.

If by means of preheating the air or by other expedients a great elevation of the temperature of ignition be secured, it has been found possible to exceed the limit at which oxygen will combine with carbon or hydrogen. The gases move separately without uniting, or if united they seem to separate. This excessively high temperature is called the " temperature of dissociation of the gases."

In the combustion or ignition of solid fuel it is probable that the first effect of the heat of the igniting flame is to distil off from the surface or render gaseous a thin external film, which gas combines with the oxygen. All ignition or inflammation takes place at the surface of large bodies whether of solid, liquid, or gaseous combustibles. Hence the imperative necessity of intimate mixture of oxygen with the combustible gas if combination is to take place in a short period of permitted contact.

Incandescence is strictly to be defined as a condition of great heat energy, accompanied by light and heat, without chemical action. True incandescence is that of the filament in an incandescent or glow electric lamp. The so-called incandescence of the glow-worm and that which appears in forms of phosphorescence and fluorescence are so called only by a permitted extension of the term. True incandescence involves the idea of light due to heat. Incandescence, however, is often extended to include the condition in which the chemical action is quite relatively slow. It is in this sense that the particles of glowing carbon in a flame-current are incandescent, or the bed of coke or carbon free from volatile

compounds is incandescent when undergoing slow combustion without appearance of flame or gas.

It is the slowness or reluctance of the combination with oxygen on the part of the incandescent particles in flame which makes perfect combustion of luminous flames a troublesome problem in some cases. If the carbon particle does not burn to gas while hot enough to unite with oxygen, it cools to black-carbon, lampblack, or soot. A current of gas carrying such solid black particles in it is called " smoke " in the engineering use of that word.

21. Spontaneous Combustion. Explosion. — While it usually requires the heat of a flame to start combustion by ignition, yet the absorption of oxygen by a body in a favorable condition for this action may be so rapid that this chemical combination will raise the temperature of a combustible up to the point at which it will burst into a flame. This action is called spontaneous combustion. Spontaneous ignition would be a better term and more exact. The condition favorable for it is the presence of a readily oxidizable body, distributed in a finely divided state over some material whereby a great surface is exposed to action by the oxygen. Oily rags and greasy waste fill this condition, and both are particularly liable to the accident. The more oxidizable the oil, the worse the danger. Vegetable oils are particularly liable to this rapid action. Coal-dust in bunkers, by reason of the oxidation of the sulphur in it, also may set fire to itself. If the heat of oxidation can be conducted off as fast as generated, spontaneous ignition is less likely to occur, but as a rule the porosity which exposes a large surface to oxidation is unfavorable to the transfer of the heat. Capillary action may also act to help the oxidizing process.

An *explosion* is a form of combustion or ignition which is practically instantaneous, or so rapid that a large volume of gas is generated and fills the volume previously occupied by the material which has been transformed into gas from some

less bulky form.   This great increase of volume forces the air in every direction, and its concussion outward or its return inward causes the report or noise.   The conditions for an explosion are the presence of combustible gas mixed with air and brought by flame at some one point to the temperature required for ignition; or the combination of gasifying solids with others rich in oxygen and ready to give it up, and the bringing of one point up to ignition by heat or friction or percussion.

A phenomenon essentially identical with an explosion appears when an atmosphere is full of a combustible dust in a finely divided state.   The fine division produces a condition analogous to that when spontaneous combustion is probable; the large surface for oxidation makes the gas from combustion form so copiously and rapidly that the ignition is practically an instantaneous gasification when a spark or flame is introduced into such an atmosphere.   Coal-dust in mines, and flour-dust in mills, are liable to this form of rapid gasification.

22. Calorific Power of a Fuel.—It will be obvious that different elements and different compounds will differ from each other in their ability to supply heat for use in a heat-engine.   The Calorific Power of a fuel is the amount of heat, expressed in thermal units, which is liberated upon the combustion of a unit of weight of the combustible material.   The calorific power of a fuel does not depend upon the rapidity of the combustion nor on the time taken in the process of absorbing the total heat of the combustion.   The temperature produced by the combustion does depend upon the rate of combustion, as will be seen in the next paragraph.

The calorific power of a compound is the sum of the calorific powers of its constituents.   If, therefore, the calorific power of the elements carbon and hydrogen have been carefully and exhaustively determined in the physical laboratory, the calorific power of a natural fuel or an artificial mix-

ture may either be computed from the percentage or weight of each constituent in the analyzed fuel, or the fuel may itself be exposed to experimental determination of its calorific power as the elements were.   This latter method is the most satisfactory.

23. Coal-calorimeters.—The calorific power of a fuel is found experimentally by causing a known weight of the fuel to burn in a closed vessel surround- ed by an observed weight of water. The number of heat-units absorbed by the rise of that weight of water through its observed range of tem- perature gives the calorific power. The apparatus used in this experi- ment is called a calorimeter or heat-measurer, and rightly.   Fig. 4 illustrates Mr. Geo. H. Barrus' ap- pliance.   To eliminate errors caused by introducing the nitrogen of at- mospheric air, oxygen gas is usually supplied to support combustion. It would be foreign to the present purpose to pursue the subject of calorimetry fully, but references to more exhaustive treatises will be found in the Appendix.   Values for the calorific power are given under the data concerning fuels (§§ 58 to 60).

BARRUS' COAL CALORIMETER
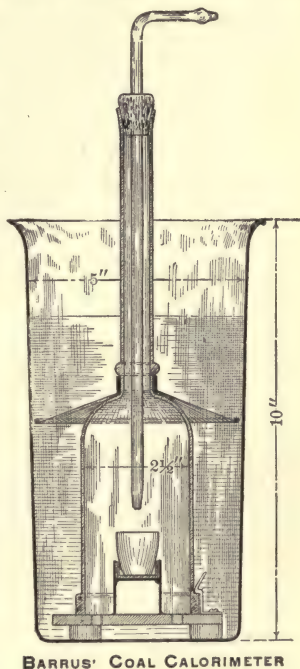
FIG. 4.

24. Air Required for Combustion of Carbon.—Since combustion is the chemical union of oxygen with the com- bustible elements, it must take place according to the laws of chemical combinations, and the weights of air for each ele- ment will be those which will furnish the oxygen weight de- manded by the relations of the atomic weights in the chemical compounds which are formed.

Atmospheric air contains oxygen and nitrogen in the following proportions, at a temperature of melting ice:

	By Weight.	By Volume.
Oxygen..................	0.236	0.213
Nitrogen................	0.764	0.787
	1.000	1.000

Whence a given quantity of air weighs $\frac{1000}{236} = 4.25$ times the weight of the oxygen which it contains, and $\frac{1000}{764} = 1.31$ times the weight of nitrogen which it contains.

By volume a given quantity of air will occupy $\frac{1000}{213} = 4.69$ times the volume of oxygen which it contains; and $\frac{1000}{787} = 1.27$ times the volume of the nitrogen which it contains.

When carbon burns to carbonic acid, which is the normal and preferred combustion process, the chemical equation for the process and result is

$$C + O_2 = CO_2,$$
$$12 + 32 = 44,$$
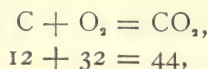
in which C is the symbol for one part by weight of carbon; O_2 is the symbol for the two parts of oxygen required to burn the carbon to carbonic acid, whose symbol is CO_2. The figures below each are the respective multiples of their atomic weights for combination; whence it appears that the oxygen weight needed will be given by the proportion:

$$\left. \begin{array}{c} \text{Weight of oxygen} \\ \text{required} \end{array} \right\} : \left\{ \begin{array}{c} \text{Weight of carbon} \\ \text{furnished} \end{array} \right. :: 32 : 12$$

or 2.66 pounds of oxygen must be furnished to burn the one pound of carbon completely. The weight of the carbonic acid, CO_2, will be the sum of the weights of carbon and oxygen, or $1 + 2.66 = 3.66$ lbs.

When the combustion is effected by supplying atmospheric air, there must be supplied from the foregoing calculation concerning atmospheric air $2.66 \times 4.25 = 11.3$ lbs. of

air. Add 1.0 lbs. of carbon. The products of the combustion will weigh 12.3 lbs. and will consist of carbonic acid and nitrogen.

Similarly the volume of air in cubic feet to burn one pound of carbon can be calculated from the weight of it. At atmospheric pressure and at the temperature of melting ice a pound of air occupies 12.39 cubic feet. Hence 11.3 pounds of air will occupy $11.3 \times 12.39 = 140$ cubic feet at 32° F., or 152 cubic feet at 62° F.

When carbon (C) burns to carbonic oxide (CO) instead of to carbonic acid (CO_2),

$$C + O = CO,$$
$$12 + 16 = 28,$$
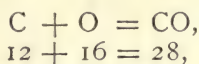
whence the oxygen is $\frac{16}{12}$ of the unit weight of the carbon, and 1.33 pounds of oxygen or $1.33 \times 4.25 = 5.65$ pounds of air are required. The products of the combustion are 2.33 pounds of carbonic oxide. The weight of air for this combustion will be $1.33 \times 4.25 = 5.65$ pounds of air, or $5.65 \times 12.39 = 70$ cubic feet of air at 32° F., or 76 at 62° F.

If the CO burns as a combustible gas to CO_2, the additional supply of air is required as in the preceding case.

25. Air Required for Combustion of Hydrogen.—Hydrogen burns to water-vapor or steam-gas, whose chemical symbol is H_2O. The chemical equation is

$$H_2 + O = H_2O,$$
$$2 + 16 = 18,$$
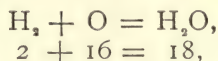
whence one pound of hydrogen requires $\frac{16}{2} = 8$ pounds of oxygen, and $8 + 1 = 9$ pounds of water-vapor result as products of the combustion, if oxygen is used alone.

Eight pounds of oxygen need $8 \times 4.25 = 34$ pounds of air, making $34 + 1 = 35$ pounds of water and nitrogen as the actual weights of the products of combustion. The volume of air for hydrogen combustion is $34 \times 12.39 = 421$ cubic feet of air at 32° F. or 457 cubic feet of air at 62° F.

26. Air Required for Combustion of Compounds.—In the burning of compounds of carbon and hydrogen each acts as though the other did not exist, and the air required is the sum of the requirements of the constituents.   Marsh-gas, for instance, known also as light carburetted hydrogen or methane, of composition CH_4, requires

$$C + O_2 = CO_2 \quad = 12 + 32 = 44$$
$$H_4 + O_2 = 2(H_2O) = \underline{\ 4 + 32} = \underline{36}$$
$$\text{Total} = 16 + 64 = 80$$

The added oxygen is four times the weight of the original gas, or one pound of gas gives five pounds of carbonic acid and water if no nitrogen is added.   Four pounds of oxygen will be furnished by $4 \times 4.25 = 17$ pounds of air at $32°$, or $17 \times 12.39 = 208$ cubic feet of air at $32°$, and giving 18 pounds of CO_2, H_2O, and N.

The proportions of the CO_2 and H_2O were respectively $\frac{44}{80}$ of the former and $\frac{36}{80}$ of the latter; or there was one part of water to 1.22 parts of carbonic acid, since

$$36 : 44 :: 1 : 1.22.$$

Similarly for olefiant gas, ethylene, C_2H_4, the equations will be

$$C_2 + O_4 = 2CO_2 = 24 + 64 = \ \ 88$$
$$H_4 + O_2 = 2H_2O = \underline{\ 4 + 32} = \underline{\ 36}$$
$$\text{Total} = 28 + 96 = 124$$

That is, for a weight of gas (28) will be required a weight of oxygen (96), or 3.43 pounds for one pound of gas, making 4.43 pounds of CO_2 and H_2O, and calling for $3.43 \times 4.25 = 14.58$ pounds of air, or $14.58 \times 12.39 = 180$ cubic feet of air, at $32°$.

The products of combustion will be $14.58 + 1 = 15.58$ pounds of CO_2, H_2O, and N, and in this combustion one part of water goes to 244 parts of carbonic acid.

If there is sulphur enough in the fuel not to be negligible,

then an additional chemical equation is required and more oxygen; S burns to SO_2, or $32 + 32 = 64$.

One pound of oxygen is required for each pound of sulphur, corresponding to 4.25 pounds of air or $12.39 \times 4.25 = 52.65$ cubic feet of air at 32° or 57 cubic feet at 62° F.

27. Combustion of an Analyzed Fuel.—The chemical analysis of a fuel gives the percentage or weight of C, H, S, and O in a pound. Hence the calculation for the weight or volume of air is identical with the foregoing, except by reason of the provision for satisfying the oxygen in the fuel itself. The investigations of Dulong and Despretz and others have shown the principle to hold, that when oxygen and hydrogen exist in a compound in the proper proportions to form water by union with each other, these constituents have no effect either in affecting the calorific power or the demand for outside oxygen for combustion. It is only the surplus hydrogen above that necessary to form water with the oxygen which need be considered; or instead of using the total per cent or weight of hydrogen, the latter is diminished by one eighth of the weight of oxygen, since one part of hydrogen by weight goes to eight weights of oxygen.

By volume

each per cent of C requires $140 \times C \div 100$ cu. ft. of air,
"   "   " H   " $421 \times H \div 100$ "   "   "
"   "   " S   " $52 \times S \div 100$ "   "   "

so that the above principle gives

$$\text{Volume of air} = \frac{140C + 421\left(H - \dfrac{O}{8}\right) + 52S}{100}.$$

By weight, for a fuel containing C and H,

$$\text{Weight of air} = 11.3C + 34\left(H - \frac{O}{8}\right).$$

This is more usually written:

$$\text{Weight of air} = 12C + 36\left(H - \frac{O}{8}\right).$$

28. Weight of Products of Combustion with an Analyzed Fuel.—Bringing together the data and results of the preceding paragraphs:

One pound of C makes $2.66 + 1 = 3.66$ pounds of CO_2;

"   "   " H   "   8 $+ 1 = 9$ "   " H_2O;

"   "   " S   "   1 $+ 1 = 2$ "   " SO_2.

Hence if C, H, and S denote the respective percentages of these elements in the fuel in question, then

$$3.66 \div 100 = .0366C = \text{weight of } CO_2,$$
$$9 \quad \div 100 = .09H \quad = \quad \text{"} \quad \text{"} \quad H_2O,$$
$$2 \quad \div 100 = .02S \quad = \quad \text{"} \quad \text{"} \quad SO_2,$$

if the combustion were in oxygen. Being effected by air, however, the weight of the products of combustion will be greatly increased, since the nitrogen weighs $\frac{764}{236}$ or 3.23 times the weight of oxygen it contains. Hence for

1 pound of C are added $2.66 \times 3.23 = 8.59$ pounds N,

1   "   " H "   "   8 $\times 3.23 = 25.84$ "   N,

1   "   " S "   "   1 $\times 3.23 = 3.23$ "   N,

1   "   " N "   "   1 $= 1$ "   N,

if the fuel analysis shows nitrogen in it. So that if these weights be reduced to percentages of each by multiplying the weight of each constituent by the above weight of N and dividing by 100, the weights of CO_2, H_2O, SO_2, and N become increased as shown in this expression:

$$
\text{Weight of Products of Combustion} =
\begin{cases}
\text{Per cent of C} \times \begin{cases} .0366 \text{ for } O_2 \\ .0859 \text{ " } N \end{cases} = .122C; \\
\text{"} \quad \text{" H} \times \begin{cases} .090 \text{ " } \frac{O}{2} \\ .258 \text{ " } N \end{cases} = .34C; \\
\text{"} \quad \text{" S} \times \begin{cases} .020 \text{ " } O_2 \\ .032 \text{ " } N \end{cases} = .052S; \\
\text{"} \quad \text{" N} \times \quad .01 \quad \text{ " } N = .010N;
\end{cases}
$$

in which C, H, S, and N in the last column are the respective percentages as given by the analysis.

29. Volume of Products of Combustion with an Analyzed Fuel.—While the foregoing calculation is most general, since the weight is independent of the temperatures of the products of combustion, the practical form of the problem is more often concerned with their volume, and this varies with the temperature. With the same conditions of pressure, if V_0 denote the volumes at the temperature of melting ice as given hitherto, and T_0 the corresponding absolute temperature, while V and T are the volume and absolute temperature corresponding to the state of the hot and expanded gases, a later discussion will show that $V_0 T = V T_0$, or the volumes will be proportional to the absolute temperatures, whence

$$V = \frac{V_0 T}{T_0}.$$

Similarly if the initial volumes be observed or taken at 62° F., the final or expanded volumes can be calculated. For example,

CO_2 at 62° occupies 8.594 cu. ft. to the pound;
H   "   "   "   190   "   "   "   "   "
SO_2 "   "   "   5.848 "   "   "   "   "
N   "   "   "   13.501 "   "   "   "   "

whence

.0366C $\times$ 8.594 = .315C   = cu. ft. of CO_2 at 62°;
.090H $\times$ 190   = 1.9H   = "   "   "   H   "   "
.02S $\times$ 5.85 = .117S = "   "   " SO_2 "   "
.0859C
.2584H $\Big\}$ N $\times$ 13.501 = $\begin{matrix} 1.16C \\ 3.49H \end{matrix} \Big\}$ = "   "   "   N   "   "

Adding and neglecting the smaller weight of SO_2, the volume at 62° F. becomes

$$V_1 = 1.475C + 5.39H,$$

and the volume V_2 at any greater temperature will be found

by multiplying the above expression by the fraction $\frac{T_2}{T_1}$, in which T_1 is the absolute temperature corresponding to 62° F., and T_2 the absolute temperature at which the volume is sought.

30. Dilution of the Products of Combustion.—Where the arrangements for the air-supply to the combustible fuel are inadequate, and where proper mixture of oxygen and the combustible gases is lacking, it may come about that the spaces in and above the fire are so filled with CO_2 or carbonic acid gas, which is not a supporter of combustion, that an excess of air has seemed to be required to dilute the excess of CO_2 and to secure a complete combustion. This difficulty has appeared at its worst with fuels containing volatile combustible gases, or which contain moisture, and which are burned with the natural draft caused by a short chimney. The excess of gas distilled off from the coal after freshly charging it upon the fire finds difficulty to get oxygen enough at temperature high enough under these conditions, when the provision for the supply of oxygen is abundant for a later period in the combustion. Water-vapor also, from damp fuels, keeps oxygen away from the combustible gases, by simple mechanical displacement, if the temperature is not high enough to dissociate the oxygen and hydrogen of which it is composed. It may act also to sweep along with it particles of carbonaceous matter, which show as smoke when unconsumed.

Hence as far back as the investigations of Prof. Johnson for the U. S. Government (1844), the principle was advocated of introducing twice as much air into a furnace as the theoretical computations demanded; or where 12 pounds of air per pound of carbon fuel were theoretically required, the engineer should arrange to introduce 24 pounds. This rule has also been followed and urged by British engineers, who were familiar with gaseous coals and other fuels.

A better understanding of the proper conditions, and attention to the matter of minutely dividing and distributing the air-supply, have reduced this excess from twice the theoretical weight to one and one half times, or 18 pounds where 12 is needed. And the best modern practice with the regular feeding of fresh fuel by mechanical stokers, and the principles and applications of forced mechanical draft in furnaces, have reduced this excess of diluent air to its lowest terms, and in many successful cases only the theoretical quantity is supplied. This feature is specially an advantage of firing with gas as fuel to be discussed hereafter. The excess of air, and especially its nitrogen, has to be heated, and by its presence lowers the average temperature prevailing at or near the fire. It may even check combustion of any gas which it can cool below the ignition-point, and modern designers are seeking to improve practice along these lines and to approach more nearly to the requirements of the chemistry of combustion. Data for weight of air and the effect of excess of air on the temperature of the fire will appear with the discussion of fuels in a succeeding chapter.

CHAPTER V.

FUELS.

31. Introductory.—A fuel has already been defined (§ 19) as a carbon or hydrogen element or a hydrocarbon compound found native or manufactured so as to be obtainable in sufficient quantities and at a low cost, so as to be used as a source of heat. Fuels are solid, liquid, and gaseous.

32. Solid Fuels. Anthracite.—The solid fuels include:

COAL
- Anthracite
- Semi-anthracite
- Bituminous
  - Dry bituminous
  - Caking bituminous
  - Long-flaming bituminous, or cannel

LIGNITE
- ASPHALT
PEAT
COKE

WOOD
- Tree-wood and slabs
- Bagasse
- Tan-bark
- Straw and stubble

CHARCOAL
ARTIFICIAL FUEL BRIQUETTES

Such fuels all seem to have been of vegetable origin, and the differences between the coals seem to have been mainly due to varying conditions during their formation in geologic periods, varying pressures after formation, and varying

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antiquity. The following table shows a relation between the
fuels on the basis of the completeness of the carbonization
of the wood fibre or cellulose:

Description.	Carbon.	Hydrogen.	Oxygen.
Wood-fibre (cellulose)	52.65	5.25	42.10
Peat..........................	60.44	5.96	33.60
Lignite..........	66.96	5.27	27.76
Lignite (brown coal)..............	74.20	5.89	19.90
Coal (bituminous)................	76.18	5.64	18.07
Coal (semi-anthracite)	90.50	5.05	4.40
Anthracite......................	92.85	3.96	3.19

From their vegetable origin the solid fuels usually contain
a proportion of incombustible mineral matter, sometimes fusi-
ble and sometimes not, which is known by the general name
of ash. The ash from wood is mainly composed of the alka-
lies: coal-ash may be iron, clay, alkaline earths, etc.

Anthracite is often called hard coal—sometimes blind coal,
or stone coal,—and consists almost entirely of fixed or free
carbon, with inorganic matter or slaty material. It was formed
at high heat and great pressure in geologic periods, and has
little if any volatile matter or hydrocarbons. It is this pres-
sure which gives it its greater hardness and density as com-
pared with a coke which has the same carbon analysis.

Its hardness and compact structure cause it to break up
or decrepitate when charged upon a hot bed of fuel, and the
small chips are liable to loss by falling into waste with the ash,
or to be carried away with strong draft of air through or over
the fire. The English and French anthracites are worse in
this respect than the American.

Anthracite is hard and lustrous, with vitreous fracture;
does not break in transportation; is not easily ignited; burns
with a short flame (unless damp) and little or no smoke;
gives an intense fire; is readily extinguished by cutting off
the air, or by cooling the fire-temperature. Its hardness and
strength make it possible to break and screen anthracite to

size without excessive loss, and in Pennsylvania **the** recognized sizes are, from the smallest downward:

Designation.	Diameter of Perforation over which Coal will pass.	Diameter of Perforation through which Coal will pass.
Dust	....	3/32 inch
Barley...............	1/16 inch	1/16 "
No. 3 buckwheat	3/32 "	3/16 "
Bird's-eye......................	1/8 "	5/16 "
No. 2 buckwheat, or rice	3/16 "	3/8 "
No. 1 buckwheat	3/8 "	9/16 "
Pea............................	9/16 "	7/8 "
Chestnut............	5/8 "	1⅛ "
Small stove......................	1 "	1¾ "
Large stove	1¾ "	2¼ "
Egg	2¼ "	2¾ "
Broken.................	2¾ "	4 "
Steamboat	4 "	7 "

What passes through a sieve of $\frac{1}{16}$ inch mesh is called waste or culm or dust. Anthracite shades off into graphite on the one side—Rhode Island anthracites are so hard as to be difficult to burn alone—and into the softer varieties on the other. The semi-anthracites have some volatile matter, break with a lamellar or conchoidal fracture, and are more readily ignited. Their flame is short, however. The analyses of typical anthracite average:

Per cent.

Fixed carbon. . . . . 90 to 94

Volatile matter { Hydrogen, 1 to 3 / Oxygen and nitrogen, 1 to 3 } . . . 3 to 7

Water. . . . . 1 to 2

Ash. . . . . 4 to 3

with a specific gravity of 1.57.

The semi-anthracites will average (Wilkesbarre):

Fixed carbon...................... 88.90

Volatile matter................... 7.68

Earthy matter or ash 3.49

with a specific gravity of 1.4.

The anthracites do their work of heating mainly by the radiant heat of the incandescent or glowing carbon. They will therefore be burned in furnaces with a large grate-area, and the fire will be as thin as consistent with having no holes in it. The use of water in ash-pits of such fires often increases the apparent flame by decomposition of the steam-gas, which cools the fire by the heat required to decompose the water, but which same heat is regained beyond the fire upon the re-composition of the dissociated gases.

American anthracites are mainly found in the eastern parts of the Allegheny Mountains and in the Rocky Mountains of Colorado.

33 Bituminous Coals. —Bituminous coals are separated from the anthracites by the possession of varying amounts of volatile matter, which distil off as gases upon the application of the heat of the fire upon which the coal is charged. The semi-anthracites or semi-bituminous coals form a wide class intermediate between those which have the characteristics of their several groups strongly marked. They have from 15 to 20 per cent of gaseous matter, a high heating or calorific power, but are of little value for making illuminating-gas by the retort process. They are valued for steam-making, because while burning with a good flame they do not give so long a flame as to be inconvenient in a boiler-setting. The following analyses of Maryland and Pennsylvania types will illustrate these properties:

Locality.	Specific Gravity.	Fixed Carbon.	Volatile Matter.	Sulphur.	Ash.
Cumberland, Md........	1.41	68.44	17.28	0.71	13.98
Blossburg, Pa..........	1.32	73.11	15.27	0.85	10.77

The true bituminous coals are softer than the anthracites, have less lustre on the fracture, break into layers or lamellæ or splints, and are the most widely distributed of the fuels.

Their fragility makes it inconvenient to sort them into sizes as can be done with anthracite, so that but three grades are usual: lump, nut, and slack. When no attempt is made to sort by size the coal is known as the "run of the mine," or is used as extracted from the bed in which it lay.

Dry bituminous, open-burning, free-burning, or furnace coal is a class of coal which can be used native in the blast-furnace, because of the absence of pitchy or " fat " material in its composition which would cause it to cake together and agglomerate into lumps upor exposure to heat.

Caking or coking coals have this property of caking together, and after the distillation from them of the 30 to 50 per cent of volatile matter which they contain the residue is a valuable coke, available for furnace or other industrial use. These usually have sulphur in them, but when free from it a valuable gas-coal is the result.

The long-flaming or cannel coals have more gas than the foregoing, but are usually lower in calorific power, and are of less industrial importance because their coke is not so valuable. They lack a certain pitchy brilliancy found in the other varieties, and are usually higher in ash.

Splint-coal is a variety of cannel with a high per cent of carbon, high calorific power, but less percentage of gas.

The bituminous coals are easily ignited, and by reason of the readily oxidizable character of the pyrites usually contained in them are liable to spontaneous ignition (§ 21) in their bunkers or bins. They form, however, the basis of successful industry in England and America and elsewhere.

Typical proximate compositions are as follows:

> Fixed carbon............ 52 to 84 per cent.
> Volatile matter.......... 48 to 12   "
> Earthy matter.......... 2 to 20   "
> Sulphur................. 1 to 3   "

Ultimate analyses of representative coals would show:

Carbon.................... 75 to 80 per cent.
Hydrogen............... 5 to 6 "
Nitrogen................ 1 to 2 "
Oxygen................. 4 to 10 "
Sulphur. .... 0.4 to 3 "
Ash or earthy matter. .... 3 to 10 "

Cannel-coals would be fairly represented by the following table:

COMPOSITION OF CANNEL-COAL.

Locality.	Specific Gravity.	Fixed Carbon.	Volatile Matter.	Earthy Matter.
Franklin, Pa......................	....	40.13	44.85	15.02
Dorton's Branch, Ky.............	1.25	55.1	42.9	2.0
Breckenridge, Ky................	....	32.0	55.7	12.3
Davis County, Ind......	1.23	42.0	52.0	6.0

34. Lignite.—The lignites are coals of more recent geologic period than the previous fuels, the process of carbonization of the wood not having proceeded so far nor under so great pressure. Lignites are brown or black. The brown lignite is sometimes of a woody texture, while the black is either woody or of a homogeneous structure with a resinous fracture. They occur in California, Utah, Colorado, Wyoming, Arizona, Alaska, New Mexico, and Oregon. Their heating capacity will be from one half to two thirds that of the older bituminous coals, and they are tender to transport. Their coke is either powdery or fibrous like that of the original wood. Lignites contain more oxygen than coal. The following table will show their composition:

COMPOSITION OF LIGNITE.

Locality.	Specific Gravity.	Fixed Carbon.	Volatile Combustible Matter	Water.	Ash.	Total Volatile Matter.	Coke.
Kentucky.....	1.201	40.00	23.00	30.00	7.00	53.00	47.00
Washington ..	....	52.85	31.75	7.00	3.00	61.25	38.75
Colorado......	1.271	41.25	46.00	3.50	9.25	50.50	49.50

35. Asphalt.—Asphalt as a fuel has a large proportion of hydrogen and burns like a tar or fat bituminous coal. It yields a porous coke, and in comparison with lignite shows the following average composition:

	Lignite.	Asphalt.
Carbon................	69 per cent.	79 per cent.
Hydrogen..............	5 "	9 "
Oxygen and nitrogen...	20 "	9 "
Ash...................	6 "	3 "
	100	100
Coke from analysis...	47 per cent.	9 per cent.
Heating power	13,108 B.T.U.	17,040 B.T.U.

36. Peat.—Peat is derived from the bituminization of mosses, grasses, and similar matter, as lignite is derived from more massive wood. It occurs in bogs, in which the upper part is turf, and peat occurs below. As piled and dried in the air after digging it contains from 25 to 30 per cent of water and from 7 to 11 per cent of ash. Regnault's standard analysis of dry peat shows:

Carbon...................	58 per cent.
Hydrogen.................	6 "
Oxygen...................	31 "
Ash......................	5 "
	100

Freshly dug peat will show 75 to 80 per cent of water. It is little used in America, but is of importance in England, Belgium, Germany, and Sweden, on account of its low cost.

A typical composition of ordinary Irish peats, both exclusive and inclusive of the moisture, which they always contain in their natural condition, would give:

<div align="center">EXCLUSIVE OF MOISTURE.</div>

Description.	Moist-ure.	C	H	O	N	S	Ash.	Coke.
Good air-dried	......	59.7	6.0	31.9		....	2.4	....
Poor air-dried......	......	59.6	4.3	29.8		....	6.3	....
Dense, from Galway	......	59.5	7.2	24.8	2.3	0.8	5.4	44.3
Averages	......	59.6	5.8	29.6		0.3	4.7	....

INCLUSIVE OF MOISTURE.

Good air-dried	24.2	45.3	4.6	24.1	......	1.8	......
Poor air-dried......	29.4	42.1	3.1	21.0	......	4.4	......
Dense, from Galway	29.3	42.0	5.1	17.5 | 1.7	0.6	3.8	31.3
Averages	27.8	43.1	4.3	21.4	0.2	3.3	......

The average composition of Irish peat, disregarding sulphur, which is seldom present, at least in quantity sufficient to have any appreciable influence, may be taken to be as given below:

AVERAGE COMPOSITION OF IRISH PEAT.

Constituents.	Perfectly Dry.	Including 25 per cent of Moisture.	Including 30 per cent of Moisture.
Carbon..............	59.0	44.0	41.2
Hydrogen	6.0	4.5	4.2
Oxygen	30.0	22.5	21.0
Nitrogen	1.25	1.0	0.8
Ash	4.0	3.0	2.8
Moisture	....	25.0	30.0

The thermal value of dry Irish peat would be (§ 22):

$$\text{Carbon.....} \quad 14{,}650 \times 0.59 = 8{,}643.5 \text{ B.T.U.}$$

$$\text{Hydrogen..} \quad 62{,}100 \times \left(.06 - \frac{30}{8}\right) = 1{,}397.25 \text{ B.T.U.}$$

$$\text{Total} = 10{,}040.75 \text{ B.T.U.}$$

37. Coke.—When a coal containing a proportion of volatile hydrocarbon or other gas is exposed to a distilling action by heat, either in a retort or an oven, the residue is called coke after the distillation is complete. It is usually dark gray in color, porous, with a slight metallic lustre, hard and brittle. It is likely to contain from 80 to 93 per cent of fixed carbon, from 17 to 5 per cent of ash, and the remainder sulphur or other impurities. The weight of coke ranges at about 66 per cent of the coal charged, with an increase in bulk of about one fourth. It will absorb moisture with avid-

ity, up to even 20 per cent. It gives a nearly smokeless combustion, with short or no flame if dry, and burns with a steady constant fire.

Modern coke-ovens condense and recover the gases from the distillation, and give most valuable by-products of the coke manufacture. The two best known are the Semet-Solvay and the Hofman-Otto, saving tar and sulphate of ammonia, and if possible using the excess of combustible gas as fuel.

Coke is less used in America as a source of motor energy, but more in metallurgy. Its absence of flame in combustion lessens its convenience for the one use and increases it for the other.

The following standard figures for the making of coke are due to Mr. A. L. Steavenson (Iron and Steel Inst. of Great Britain):

Element.	Original Analysis of Coal.	Loss in Coking.	Yield of Coke.
Oxygen..................	6.7	16.6	....
Carbon	84.9	68.1	96.2
Hydrogen................	4.5	11.2	....
Nitrogen	1.0	2.5	....
Sulphur	0.6	1.6	....
Ash	2.0	....	3.8

38. Wood.—In the older countries and the more thickly settled parts of the new, wood is becoming less and less used as a source of heat energy, by reason of its growing scarcity and the proper opposition to deforestation for the sake of the country as a whole. It is, however, of importance still as refuse in chips and dust from a wood-working process of manufacture, and for the disposition of stalks or similar annual product of tillage.

Fire-wood or slabs may be either of the soft woods, such as pine, birch, or poplar, or the hard woods, like oak, hickory, ash, elm, or beach. Ash is small in woods and varies from

1 to 5 per cent. Fresh-cut fire-wood usually holds about 40 per cent of moisture, which upon air-drying for several months will diminish to 15 to 25 per cent. If dried in kilns or ovens and exposed afterward to the air, the wood absorbs water rapidly in the first few days—perhaps 5 per cent in the first three—and thereafter will absorb slowly till the normal percentage of dry wood is reached, and this will fluctuate according to atmospheric conditions. Various tree-woods are much alike chemically, averaging as follows:

Kind of Wood.	C	H	O	N	Ash.	Moisture.
Beech...............	49.36	6.01	42.69	0.91	1.06	....
Oak................	49.64	5.92	41.16	1.29	1.97	....
Birch...............	50.20	6.20	41.62	1.15	0.81	....
Poplar..............	49.37	6.21	41.60	0.96	1.86	....
Willow	49.96	5.96	39.56	0.96	3.37	....
Averages, kiln-dry	50.0	6.0	41.0	1.00	2.0	....
Average, 25% water	37.5	4.5	30.75	0.75	1.5	25

The heating power of wood is usually called 7838 B.T.U. dry or 5879 when wet. This would be about four tenths that of an equal weight of coal, or in other words, $2\frac{1}{2}$ pounds of wood are equivalent to one pound of coal. In substituting wood for coal as a fuel, the furnace usually has to be enlarged, principally in its height, perhaps, so that it may carry an equal weight of combustible at any one time as it formerly did with coal. Wood, like the gaseous coals, distils off volatile or tarry matters, which are often sticky and brown in color, and are combustible if a high enough temperature can be commanded. These do not constitute a true smoke as heretofore defined (§ 20), but are often judged to be one, and should be as avoidable.

When sawdust from manufacturing processes is to be used as a fuel, the conditions of intimate mixture of the oxygen required for the combustion have to be secured by special forms of grate, and usually by a forced draft (§§ 24 to 26, and

86 to 88). Hollow grate-bars form a usual method of meeting this requirement (Gordon's or Gadey's), with a fan forcing
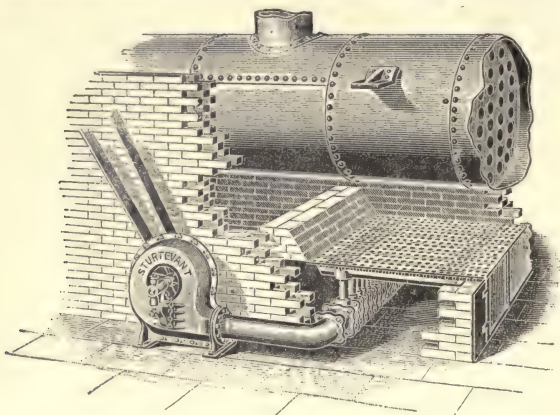
FIG. 5.

air into the cored passages and out and up through the fuel-bed (Fig. 5).

39. Bagasse, Straw, Tan-bark.—The three most usual forms of woody fibre used as refuse for fuel are the residue from crushing the sugar-cane to press out its juice, which is called bagasse; the stalks or stubble from cotton, wheat, or barley harvests; and the spent bark from tan-pits out of which the tan-liquor has dissolved the desired acids and left the woody fibre behind. The accepted analyses of tropical cane and the resulting bagasse after crushing give:

	Cane.	66% Bagasse.	70% Bagasse.	72% Bagasse.
Woody fibre..........	12.5	37	40	45
Water...............	73.4	53	50	46
Combustible salts....	14.1	10	10	9
	100.0	100	100	100

The figures 66, 70, and 72 refer to the proportion of mill extraction of the juice from the cane.

Dry Louisiana bagasse will analyze:

Constituents.	Percentage.
Volatile matter...................	81.37
Fixed carbon....................	14.26
Ash...........................	4.6

If the woody fibre contains 50 per cent of carbon, and the combustible salts 42 per cent, as has been found, then the calorific power of bagasse would be about 1200 B.T.U.; or one pound of coal equals 5 or 6 pounds of wet bagasse, or $2\frac{1}{2}$ to 3 pounds if dry. In burning it, it is fed continuously into a very hot fire-brick chamber, and generous space must be provided to take care of the volume of steam and gas generated.

Straw is composed in its ordinary and air-dried condition as follows:

	Dry Wheat Straw. Russia.	Wheat Straw. Head.	Barley Straw.	Mean.
Carbon.............	46.1	35.86	36.27	36.00
Hydrogen...........	5.6	5.01	5.07	5.00
Oxygen	43.7	37.68	38.26	38.00
Nitrogen	0.42	0.45	.40	.425
Ash	4.18	5.00	4.50	4.75
Water.............	....	16.00	15.50	15.75
	100.00	100.00	100.00	100.00

The calorific power of such straw would prove to be 8144 B.T.U., or 1 pound of coal equals $2\frac{3}{4}$ to $3\frac{3}{4}$ pounds of cotton-stalks or straw.

Tan-bark containing 30 per cent of water has a calorific power of a little over 4200 B.T.U. and contains 15 per cent of ash. That is, one pound of coal equals $2\frac{1}{8}$ to 3 pounds of dry tan-bark, or 6 to 8 pounds if the latter is wet. The evaporative capacity of tan-bark, expressed in pounds of water evaporated from and at 212°, is given as follows:

	Perfectly Dry.	With 30 per cent of Moisture.
Water supplied at 62°....	5.46 pounds	3.84 pounds
Water supplied at 212°....	6.31 pounds	4.44 pounds

The conditions of success in burning tan, as is the case with all wet fuel, consist in completely surrounding it with heated surfaces and burning fuel so that it may be rapidly dried, and then so arranging the apparatus that thorough combustion may be secured. Here again, as with sawdust, the hollow grate-bar and forced-blast systems in combination with the hot fire-brick furnace seem to meet the conditions most satisfactorily. Fig. 6 shows the overhead-hopper feed-
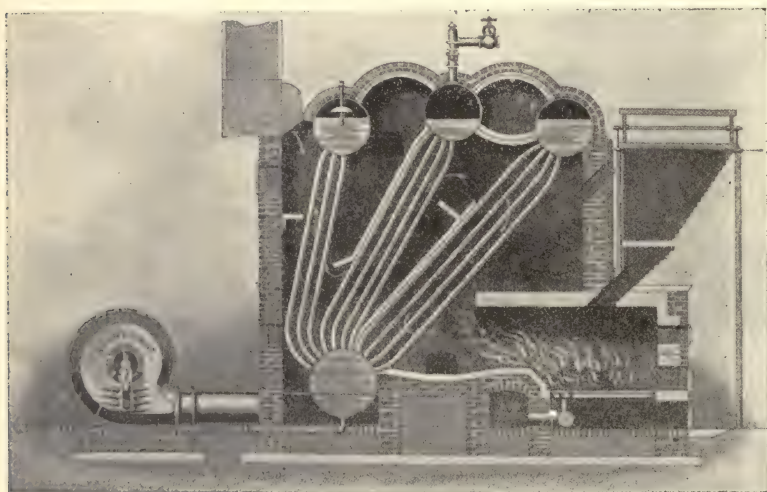
Fig. 6.

ing plan to secure automatic stoking of the furnace (§ 90), as applied to a water-tube boiler with the forced current of air from a fan entering the fuel-bed from below, and the furnace is arched over with fire-brick, which becomes very hot and secures the necessary conditions.

40. **Charcoal.**—Charcoal is the product of distillation from wood to expel its volatile constituents as coke is produced

from bituminous coal. It is done in heaps or pits or ovens at about 600° to 800° F. So much of the heat-making properties of native wood are expelled in the process of making charcoal that it is of little moment for motive-power purposes. It is easily ignited and burns with a flameless and smokeless incandescence. Charcoal has its quality improved as the temperature of its distillation is increased. The results of this process when applied to black alder, previously dried at about 300°, are as follows:

COMPOSITION OF CHARCOAL PRODUCED AT VARIOUS TEMPERATURES.

Temperature of Carbonization.	Constituents of the Solid Product.				
	Carbon.	Hydrogen.	Oxygen.	Nitrogen and Loss.	Ash.
302° F.	47.51	6.12	46.29	0.08	47.51
392	51.82	3.99	43.98	0.23	39.88
482	65.59	4.81	28.97	0.63	32.98
572	73.24	4.25	21.96	0.57	24.61
662	76.64	4.14	18.44	0.61	22.42
810	81.64	4.96	15.24	1.61	15.40
1873	81.97	2.30	14.15	1.60	15.30

Peat charcoal, produced by the carbonization of ordinary air-dried peat, is very friable and porous, and extremely difficult to handle without reducing it to very small particles almost powdery in their character. Although it is easily ignited and burns readily, its physical characteristics are such as to prevent its general use.

41. Artificial or Patent Fuels.—Combustible materials in the form of dust or grains which in that state are ill adapted to convenient use for heat-making can be made into a practicable and salable fuel by agglomerating such loose particles into blocks or bricks by means of some tarry or bituminous compound such as pitch, resin, or even glue. The fine dust or grains would clog a fire or be lost in ashes or by the draft, and would burn too rapidly and unsteadily if fed irregularly

or by hand. By moulding into briquettes the combustion is made gradual and from their outer surfaces inward, and the larger mass is not so readily extinguished. This plan is much used in Belgium and in some other places in Europe, and has been tried in America as an expedient to work over the heaps of waste coal in the dumps of mines. Mechanical stoking with proper travelling grates has been found a more successful method of using such material.

42. Liquid Fuel. Petroleum.—The second great class of fuels (§ 31) are those which can be supplied in a liquid state and burned to a gas with the consequent liberation of stored heat. These liquid fuels are hydrocarbons and are called oils. Oils of animal origin are now supplied to such a limited extent as scarcely to deserve consideration, and the cost of extracting vegetable oils from the seeds or other products which carry it preclude the use of such oils for fuel. Hence the mineral oil, or petroleum, is the principal source of heat from liquids, either in its crude form as it comes native from the oil-well, or after a part of the constituents of natural oil have been eliminated by the refining process. The average composition of crude petroleum is usually given as:

	From	To	Average.
Carbon....................	82	87.1	85
Hydrogen..................	11.2	14.8	13
Oxygen and impurities.....	0.5	5.7	2
			100

Its specific gravity is from 0.79 to 0.82. Lima oil from the Ohio wells is of a dark green color, is quite fluid and volatile, and has a disagreeable odor. Its volatility makes it flame easily, and give off an explosive vapor in a confined space. These two properties have resulted in restrictions upon its use in many cities; the health boards object to the odor, and the fire departments to the danger of fire from explosions. Hence the refining companies have introduced what is called fuel-oil. This is the residue after a part of the

fractional distillation process has been completed. A tabular summary of this process is as follows:

No.	Temperature Fahrenheit.	Distillate.	Probable Per Cent.	Specific Gravity.	Flashing-point.
1	113	Rhigolene } Chymogene }	traces	.590 to .625	....
2	113 to 140				
3	140 to 158	Gasolene	1.5	.636 to .657	....
4	158 to 248	Benzine, naphtha C	10.0	.680 to .700	14
		" " B	2.5	.714 to .718	....
5	248 to 347 {	" " A	2	.725 to .737	32
		Polishing-oils	....	....	....
6	338 +	Kerosene	50	.802 to .820	100 to 122
7	482	Lubricating-oil	15	.850 to .915	230
8	....	Paraffine wax	2	....	....
9	....	Residuum and loss	16	....	....

The distillation for fuel-oil is stopped after the kerosene has been obtained. In many refineries only the three products of crude naphtha, burning oil, or kerosene, and the distillate are recognized, the latter being the fuel-oil. Its average specific gravity is about .818 or 40° Baumé at 60° F., so that a gallon weighs 7.3 pounds, as against 6.81 pounds for the crude oil. It flashes at 218° F., or just above the boiling-point of water. It is thick in consistency. The calorific power of crude oil is from 20,000 to 21,000 British thermal units, and that of the fuel-oil is from 17,000 to 19,000 heat-units. Fuel-oil is called "astatki" by the Russians. Thos. Urquhart of Russia, in considering the use of petroleum for locomotives, gives the following table of the theoretical evap-

Fuel.	Specific Gravity at 32° F., Water = 1.000.	Chemical Composition.			Heating-power, British Thermal Units.	Theoret. Evap., Lbs. Water per Lb. Fuel, from and at 212° F.
		C.	H.	O.		
Penna. heavy crude oil..	0.886	84.9	13.7	1.4	20,736	21.48
Caucasian light crude oil	0.884	86.3	13.6	0.1	22,027	22.79
" heavy " "	0.938	86.6	12.3	1.1	20,138	20.85
Petroleum refuse.,......	0.928	87.1	11.7	1.2	19,832	20.53
Good English coal, mean of 98 samples....	1.380	80.0	5.0	8.0	14,112	14.61

orative power of petroleum in comparison with that of coal, as determined by Messrs. Favre and Silbermann.

The further details of refining for elimination of coloring matter, and the steps of acid and alkaline agitation, are aside from the present purpose.

43. Kerosene.—Kerosene has already been referred to as the " burning oil " or No. 6 in the process of fractional distillation of petroleum. Usually $3\frac{1}{2}$ parts of crude oil render one part of kerosene.

The heat of combustion ranges between 27,000 and 28,000 B.T.U. The quicker the distillation the poorer the product, albeit more abundant; but the more abundant the lighter elements the less safe is the kerosene.

The flashing-points at which an ignitible vapor is given off by heating will range between 115° and 125° F.; the oil will itself ignite and burn when heated to 130° to 140° F. This is called its burning-point. Besides its use as a lamp-oil, kerosene is used in certain forms of oil-engine to supply the heat for motive power. The limited use of the more volatile petroleum liquids will be referred to in Chapter XX (§§ 297 to 299).

44. Alcohols.—There are two kinds of alcohol used in the arts and as sources of heat: methylic alcohol or wood-alcohol, which has the chemical symbol CH_4O, and ethyl alcohol, the ordinary form, which is represented by C_2H_6O.

Wood-alcohol is formed by dry distillation of wood in iron retorts (usually horizontal) at a heat not above 900° F. It has a strong characteristic odor and boils at 150° F. It would be a most popular source of heat in many places where corn is abundant if there were no restrictions upon its manufacture.

Ethyl alcohol is obtained by distillation from the fermented infusions of the cereal grains, which contain either sugar or starch. It has a specific gravity of 0.792 and boils

at 173° F., but will freeze only at 200° below zero when pure. It expands 3⅛ times as much as water between 32° and 173° F.

Hydrated alcohols contain water ranging from 50 per cent by volume (proof spirits) to 93 per cent (cologne spirits). The affinity for water is very strong.

Pure alcohol is very inflammable and burns with a pale-blue smokeless flame. Its calorific power is about 28,500 B.T.U., which runs down to 12,000 with greater hydration.

45. Liquid-fuel Furnaces.—The heat may be derived from burning oil by setting fire to it in the presence of oxygen while the oil is a liquid, or is a finely divided mist or vapor of liquid particles in a current or stream of air or steam, or the liquid may be made into a gas and then ignited.

The heat of combustion may furthermore be utilized directly in a motor cylinder (the oil-engine, Chapter XX), or the heat may be used as the heat of combustion of solid fuels is, in a furnace or fire-box from which it is transferred by a medium to the cylinder.

The American methods have been the vapor or gas systems exclusively. The liquid systems are Russian or Indian in the main. Four general plans have been tried. First, bowl- or pan-furnaces, in which the liquid oil was delivered through a series of pipes into shallow vessels in the fire-box, and burned from their surface. This plan is old and primitive, and gave difficulty from smoke because air-supply was difficult. Second, step-furnaces, in which a series of shallow troughs was arranged across the furnace in steps, and the oil fed into these troughs from above overflowed into the one below and met the air for combustion in flat strata between the steps. This answered for stationary conditions. Third, drop-furnaces had the oil fed in thin streams from many pipes over a grooved plate, whereon it met the air and was burned. Fourth, wick- or oozing-furnaces, where the liquid

oil is made to rise up from below at many points through a
layer of an incombustible substance like lime or asbestos or
pumice-stone, or even sand. The oil divides itself among
the interstices and burns from the top surface, where it meets
the air as a lamp-flame from a wick. The difficulty is the
certain clogging of the capillary surfaces by the heavy resid-
ual matters, which are sticky and refuse to come to the top
and burn

46. Oil-vapor Burners.—The most widely extended and
successful system for generating heat from oil-fuel is to com-
minute or atomize the oil by a current of air or steam which
blows across or through it under pressure in an appliance or
apparatus which is called a burner. The oil is drawn into
the current of air or steam by induction and is torn into
shreds or drops by the high velocity through a small opening,
so that it enters as a mist into the fire-box, and ignites by
flame or heat already there, all through the saturated atmos-
phere, forming a volume of flame of great intensity and heat-
ing effect. It is rarely or never wise to have the oil flow by
gravity to the burner, since the flow of oil should positively
cease when the inducing current of air or steam is shut off,
and to diminish the fire danger.

The burner is usually a modification of the injector in
principle. Its outlet is either a slit or a nozzle. In the slit
sprinklers the orifice is divided by a thin partition, the oil
coming on one side of it and the air on the other. When the
slit is horizontal, as has been usual, the oil is above and the
air or steam blows through the film of oil. Sometimes the
partition has grooves, so that oil passes in threads. Slit
sprinklers are wasteful of oil and of steam as compared with
the other types. They will call for 6.6 pounds of oil per
H.P. per hour, and will use from 6 to 8 per cent of the
steam which is made by the boiler for which the burner sup-
plies necessary heat. They also are liable to become clogged

with the solid matters of the oil residue, and to become blown out by their own operation.

Nozzle sprinklers have the air and oil concentric in a converging tube—the air usually in the middle. The air- or steam-nozzle is usually adjustable within the oil-tube, so as to vary the flow of oil, and to permit cleansing of the oil-tube by retracting the air-nozzle. Such burners will require from 4.4 to 5.5 pounds of oil per H.P. per hour, and will consume from 4 to 6 per cent of the steam which the burner will make. When there is no nozzle adjustment the sprinkler will be called a pipe burner rather than a nozzle sprinkler. Some forms of burner aspirate a current of air first by a steam-jet, and then the combined air- and steam-current aspirate and
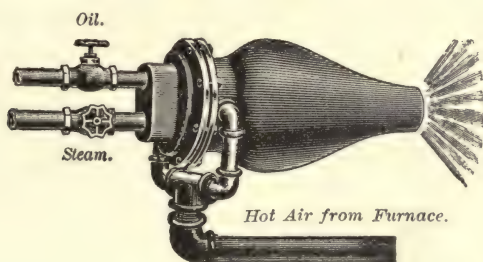
Fig. 7.

atomize the oil. The jet of oil-vapor should impinge upon fire-brick or similar refractory material, which becomes incandescent and keeps the flaming vapor alight. Metal surfaces are eroded rapidly from the intense heat when the flame impinges upon them. The fire is started by an igniter of chips or waste soaked with kerosene; but after becoming well started the mass of fire-brick will start the flame anew after the oil has been shut off for a while. There are several designs of oil-burner, known by their manufacturers' names. (Figs. 7 and 8 illustrate types.)

With respect to the use of air or steam for the inducing means to draw up and atomize the oil, it may be said on behalf of steam that it requires no air-compressing plant to bring it up to pressure for use under boilers, and there is not introduced into the flame a mass of inert nitrogen which must be heated at the expense of the oil-fuel, and acts to cool it. Steam is hot, furthermore, when it enters the flame,
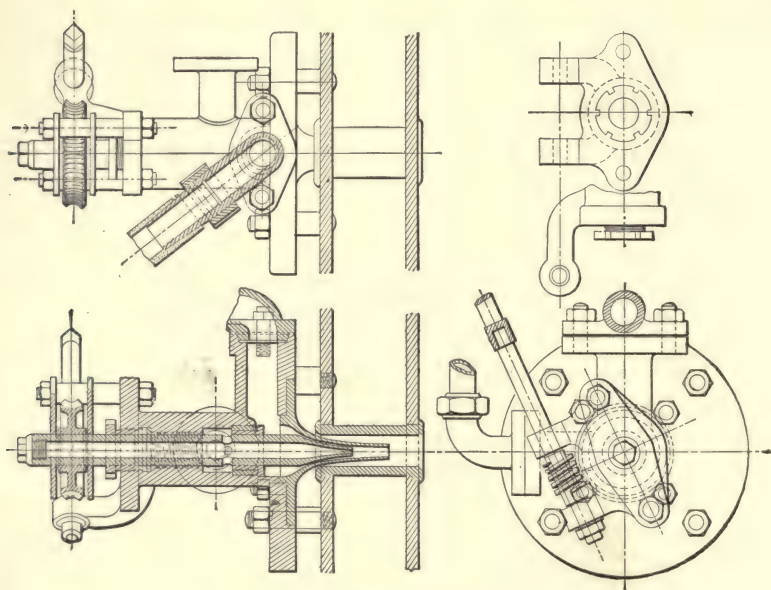
FIG. 8.

and may be superheated. On the other hand, air must be introduced for combustion, and it is best to introduce it as the spraying and subdividing medium; steam dilutes the burning gas if it is not dissociated; and if it is, the heat of dissociation is lost unless the temperature is high enough for recombination.

47. Oil-gas Systems.—The third group of methods for getting heat from oil is the plan of making the oil into a gas by heating it in a proper retort, into which also superheated steam is introduced, at a temperature above 300° F. The Archer process is one of the best known of its class, and has the steam and oil enter a cylindrical retort or thermogen in such a way as to secure an intimate mixture of the constituents. The plant at the outset involves more than the burner or vapor plans, and the retort connections are liable to clogging from the residues of the oil. Sometimes, however, by the finer subdivision and better access for air which gas-firing permits, the gas systems burn less oil for a given heating effect than the vapor-furnaces.

A form of oil-gas is much used wherein the gas is made by forcing air under pressure through a liquid hydrocarbon, such as gasoline. The carbureter is placed between the pressure-tank and the burner, and the air on its way to the burner picks up enough carbon to form an illuminating-gas. This method has been considerably applied for railway-car lighting. When so used, the air is delivered to the carbureter from the air-brake tanks, passing through a spiral coil of fine copper pipe in the chimney of the lamp. The carbureter contains a gasoline of about 88° Baumé, absorbed in a tightly compressed cotton wicking.

The Pintsch oil-gas (Julius Pintsch, Berlin, 1871) is a true or fixed gas made in retorts by vaporization of crude petroleum. From 70 to 85 cubic feet of a 50- to 60-candlepower gas results from distillation of one gallon of oil. For railroad use this gas is compressed in tanks, to 150 pounds pressure or more, which supply the smaller gas-tanks under the cars. Each of these car-tanks will hold enough for from two to six days' travel. The gas is rich in illuminating properties and does not lose so much of its illuminating power by compression as a coal-gas. A small deposit of hydrocarbon is found in the bottom of the storage tanks, however.

48. Advantages of Oil-fuel.—Oil or liquid fuel offers many attractive advantages over the solid fuels. Many of these are those incidental to mechanical firing, to which oil lends itself easily, but besides these there are many others of its own.

Mechanical handling of oil by pumps or aspirating burners gives the following advantages:

1. Economy of labor. One fireman by handling the necessary valves can manage eight to ten or more boilers of 100 horse-power each. With hand-firing of coal, one man can never manage more than four such boilers.

2. No ashes, and their attendant labor and possible cost.

Economy and convenience in oil-firing result from:

3. No waste of fuel in ashes and cleaning of fires

4. No waste of fuel in banking fires overnight.

5. No opening of furnace-doors for firing or cleaning. This is easier upon the brick-work of the setting, and on the metal of the boiler, by diminishing strains of sudden contraction.

6. No injury from firing-tools in fire-boxes.

7. No sparks pass out from a chimney, to set fire to combustibles outside.

8. Absence of dust and ashes in fire-room and adjoining engine-room.

9. Wide range of controllability of fire, not only within the limits of ordinary consumption, but beyond these. The fire is put out when demand for heat stops ; an excessive demand for heat can be met by unusually great supply of oil. With solid fuel, a charge once made must burn itself out. In boilers, safety-valve waste is diminished.

10. The greater calorific power of oil, and its controlled combustion, enable more energy to be gotten from a plant whose capacity has been calculated upon a solid fuel basis.

11. Smokeless combustion is more easily secured, and there is diminished loss of unburned carbon.

12. Lower temperatures of fire-rooms, and lessened physical strain upon firemen.

13. Absence of sulphur to corrode metal.

14. Fires easily started.

15. Economy of stowage and carriage of oil as compared with solid fuel.

16. Economy of fuel-stations for navy or locomotive practice.

17. No grates are required.

Usual relations of oil to coal as fuel give 1 pound of oil to $1\frac{3}{4}$ pounds of coal; or 1 gallon of oil equals 6.5 to 6.7 pounds of oil and will compare to 12 pounds of coal; or 190 gallons of oil will equal a long ton of coal of 2240 pounds.

49. Disadvantages of Oil-fuel.—There are objections to oil as a dependence for a source of heat.

1. The use of crude oil with the volatile constituents in it is opposed by the health ordinances of some cities. In others the fire or insurance ordinances permit the use of oil only if the oil-tank is below ground, or so placed that it cannot flow out of its reservoir and carry flame to other buildings in case of conflagration.

2. The vapor from crude oil is ill-smelling and makes an explosive mixture with air. It vaporizes even at low temperatures.

3. If fuel-oil must be used, it is usually more costly than coal in most places. The problem is really to get the most heat-units for a unit of value. If the quotient of the calorific power of oil per pound divided by its price per pound at any point is greater than the same quotient for solid fuel, the oil is the cheaper.

4. The total oil-production of the world would supply but a small portion of the demand for heat as a source of energy. This would immediately affect the price of oil, if any large number of consumers were to decide to use oil.

5. Most of the spray burners make an objectionable roaring noise.

6. The surfaces exposed to an oil-flame usually become coated with a deposit of residue from the burning oil.

7. Oil creeps past valves and leaks in a way which is annoying and may be dangerous.

8. Explosions occur from the flame blowing out, and igniting again with dangerous combinations of oil-vapor and air.

9. Auxiliary apparatus in the way of a source of steam or compressed air is required for the burners; in starting, there must be a supply available of air or steam from a boiler or reservoir.

50. Gaseous Fuels. General.—It is one of the most tenable theories of the combustion of solid and liquid fuels, that the effect of the igniting heat is to gasify the carbon or hydrocabron on the surface, and that chemical union with oxygen takes place when both are gases, the carbon gas being in a nascent state. If this view is sound, there are advantages connected with the plan of making gas on a large scale artificially, or in using natural gas as a fuel. Gas-firing offers the same advantages as a principle as those which underlie the use of oil in the matter of mechanical handling, control, cleanliness, and convenience. Gas-firing, further, requires less excess of air for combustion—or none; and when gas can be used as a source of heat energy direct in the cylinder, the advantages are introduced which follow from avoiding some of the necessities for losses which are introduced in other systems.

Gas-fuel may be natural gas or a manufactured article. Manufactured or artificial gas may be producer-gas, water-gas, or illuminating-gas.

51. Natural Gas.—In certain parts of America, notably in Pennsylvania, Ohio, and Indiana, large accumulations of a natural fuel-gas are found in subterranean cavities or strata, which can be reached by wells. This gas is either a stored

product of a previous distillation, or else is a product of a process still in operation. It is usually under considerable pressure at the wells, so that it can be piped to industrial centres without too great losses, or artificial pressure may be secured by proper gas-pumps.

Various districts give varying constitution of gas and hence varying calorific power ; around Pittsburg, Pa., I pound of coal is considered to be the equivalent of from $7\frac{1}{2}$ to $12\frac{1}{4}$ cubic feet of gas. The following tables give some analyses:

VARIATION IN COMPOSITON OF NATURAL GAS.

Constituents.	1	2	3	4	5	6
Marsh-gas	57.85	75.16	72.18	65.25	60.70	49.58
Hydrogen.............	9.64	14.45	20.02	26.16	29.03	35.92
Ethylic hydride........	5.20	4.80	3.60	5.50	7.92	12.30
Olefiant gas...........	0.80	0.60	0.70	0.80	0.98	0.60
Oxygen.	2.10	1.20	1.10	0.80	0.78	0.80
Carbonic oxide........	1.00	0.30	1.00	0.80	0.58	0.40
Carbonic acid..........	0.00	0.30	0.80	0.60	0.00	0.40
Nitrogen..............	23.41	2.89	0.00	0.00	0.00	0.00

Analyses from various wells in Indiana and Ohio indicate the composition to be as follows:

COMPOSITION OF NATURAL GAS FROM OHIO AND INDIANA.

Constituents.	Ohio.			Indiana.			
	Fostoria	Findlay	St. Mary's.	Muncie.	Anderson.	Kokomo.	Marion.
Hydrogen...............	1.89	1.64	1.94	2.35	1.86	1.42	1.20
Marsh-gas	92.84	93.35	93.85	92.67	93.07	94.16	93.57
Olefiant gas.............	0.20	0.35	0.20	0.25	0.47	0.30	0.15
Carbonic oxide	0.55	0.41	0.44	0.45	0.73	0.55	0.60
Carbonic acid.	0.20	0.25	0.23	0.25	0.26	0.29	0.30
Oxygen.................	0.35	0.39	0.35	0.35	0.42	0.30	0.55
Nitrogen................	3.82	3.41	2.98	3.53	3.02	2.80	3.42
Hydrogen sulphide.......	0.15	0.20	0.21	0.15	0.15	0.18	0.20

The Indiana gas weighs 0.045 pound per cubic foot.

52. Producer-gas.—Gas made by distilling bituminous or anthracite coal in a closed furnace, using part of its own heat of combustion to effect the chemical reactions, is often called producer-gas, from the name given to the gas-generator.   A thick bed of fuel rests upon properly constructed grates, and air or steam or both is forced from below the grates up through the bed of fuel.   The first combustion is to carbonic acid (CO_2) with air alone, or to CO_2 and hydrogen if steam is used also.   This carbonic acid gas, meeting the layers of carbon above where no free oxygen reaches, is decomposed by the carbon into two units of carbonic oxide (CO), which with the hydrogen passes up through the bed of fuel and outwards through a proper pipe to the place where it may meet the required oxygen and be burned at the point desired. Early producers of the Siemens type, operating with open ash-pits and no pressure below the grates, lost much of their possible effectiveness in the cooling of the gases after leaving the producer.   This loss is estimated at 30 per cent.   To blow with air alone is to introduce inert nitrogen which dilutes the gas and lowers its calorific power.   On account of the loss of heat in the producer itself in the distilling process, and some loss in the dissociation of the water, which is not all recovered, producer-gas usually carries only 87 per cent of the calorific energy of the carbon.   Some loss in unreduced CO_2 must be allowed for, and the cost of making the steam used.   Eighty-two per cent is a more usual figure when anthracite is used as fuel instead of bituminous coal.   Much inferior grades of fuel can be used in the producer than could be used direct, however.

If an analysis of 85 per cent of solid carbon be assumed for an anthracite stock, with 5 per cent of volatile hydrocarbons and 10 per cent of ash, and the further assumption be made of a combustion of 80 pounds to CO_2, and 5 pounds of CO, the following calculated statement of process, products, and resulting energy may be agreed to:

Process.	Products.		
	Pounds.	Cubic Feet.	Anal. by Vol.
80 lbs. C burned to...................CO	186.66	2529.24	33.4
5 lbs. C burned to.................CO₂	18.33	157.64	2.0
5 lbs. vol. HC (distilled).....	5.00	116.60	1.6
120 lbs. oxygen are required, of which 30			
lbs. from H₂O liberate...... ......H	3.75	712.50	9.4
90 lbs. from air are associated with...N	301.05	4064.17	53.6
	514.79	7580.15	100.0

Energy in the above gas obtained from 100 pounds anthracite:

$$186.66 \text{ lbs. } CO \ldots\ldots\ldots\ldots 807,304 \text{ heat-units.}$$
$$5.00 \text{ `` } CH_4 \ldots\ldots \ldots\ldots 117,500 \text{ ``}$$
$$3.75 \text{ `` } H \ldots\ldots \ldots\ldots\ldots 232,500 \text{ ``}$$
$$1,157,304 \text{ ``}$$

Total energy in gas per lb...... 2,248 ``
`` `` `` 100 lbs. of coal. 1,349,500 ``
Efficiency of the conversion............ 86 per cent.

If the gas-stock be a bituminous coal with 55 per cent of fixed carbon, 32 per cent of volatile matter, and 13 per cent of ash, and the calorific power of the hydrocarbons be taken at 20,000 heat-units to the pound, the following table results under the same assumptions:

Process.	Products.		
	Pounds.	Cubic Feet.	Anal. by Vol.
50 lbs. C burned to..................CO	116.66	1580.7	27.8
5 lbs. C burned to..................CO₂	18.33	157.6	2.7
32 lbs. vol. HC (distilled).....	32.00	746.2	13.2
80 lbs. O are required, of which 20 lbs.,			
derived from H₂O, liberate........H	2.5	475.0	8.3
60 lbs. O, derived from air, are associated			
with..............................N	200.70	2709.4	47.8
	370.19	5668.9	99.8

Energy in 116.66 lbs. CO...... 504,554 heat-units.
`` `` 32.00 lbs. vol. HC... 640,000 ``
`` `` 2.50 lbs. H........ 155,000 ``
1,299,554 ``
Energy in coal................ 1,437,500 ``
Per cent of energy delivered in gas.......... 90.0
Heat-units in 1 lb. of gas..................... 3484

Fig. 9 illustrates the old type of Siemens producer without artificial blast, and Fig. 10 the more modern Taylor producer with forced steam-blast and revolving grates.
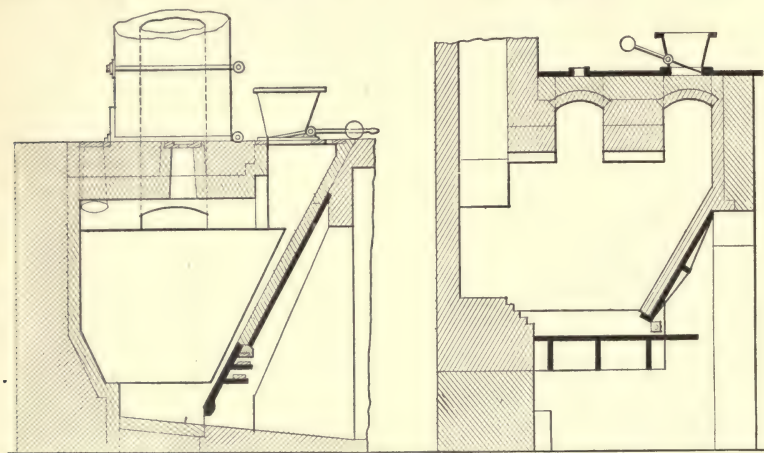
FIG. 9.

Ordinary producer-gas has a calorific value of 110 B.T.U. per cubic foot.

53. Water-gas.—A great deal of gas for illuminating and power purposes is now made by the process of intermittent and alternate blowing of air and steam through a thick bed of fuel in a cylindrical producer of boiler-plate lined with refractory material. The fuel is blown by air from below until it becomes highly incandescent; the producer may be open at the top, and waste the lean carbonic oxide which comes off from the top, or the latter can be caught and used. After blowing with air as long as necessary, the air is shut off, and steam is similarly blown from below, with the producer closed except at its delivery to a gas-holder. The steam is dissociated by the incandescent carbon into hydrogen and oxygen, and the latter unites with the carbon as in the air-producer, to be reduced to carbonic oxide. The hydrogen passes out without further chemical reaction. This process
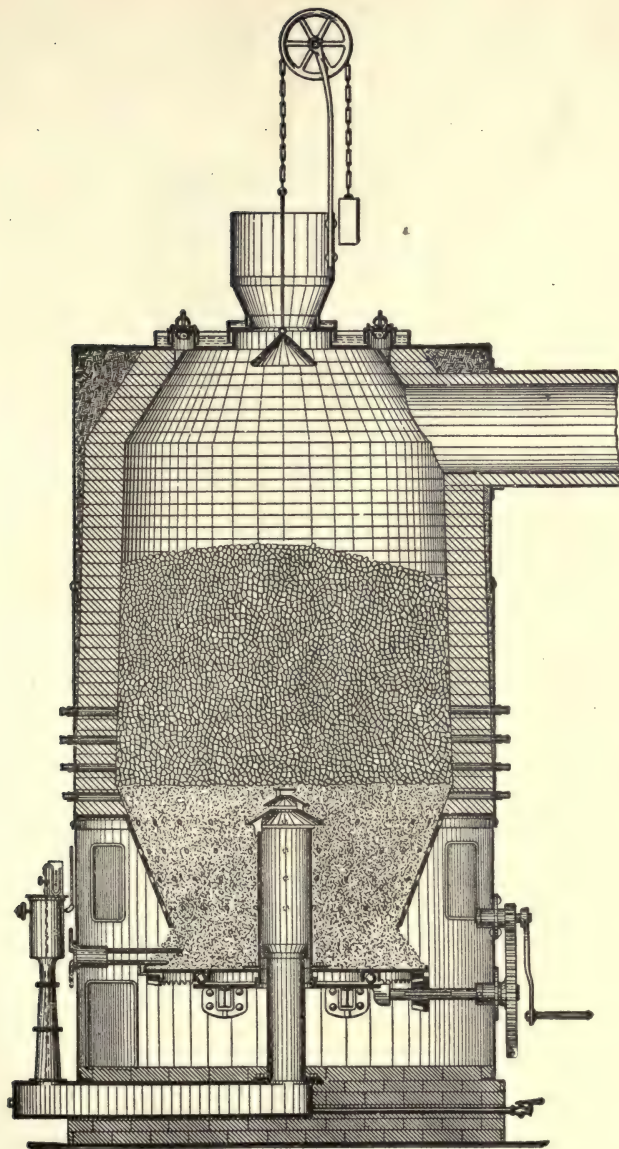
A

Fig. 10a.

was introduced in 1874 by Mr. T. S. C. Lowe, and is often known generally as the Lowe process. For illuminating purposes this fuel gas is more highly carburetted by sprays of hydrocarbon vapors (such as naphtha or similar petroleum
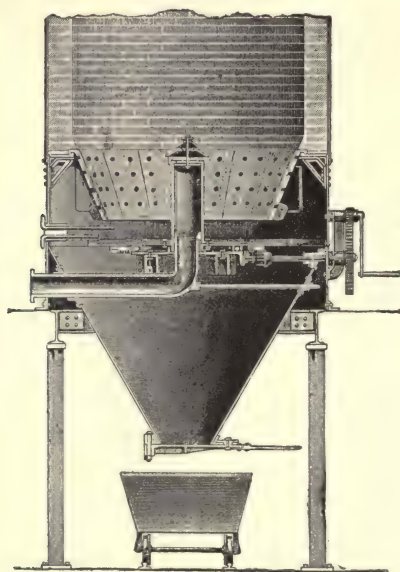
Fig. 10b.

products) which are made a fixed gas by later heating in a superheater.

Fig. 11 illustrates what is called in England the Dowson gas-producer, which belongs to this class. Its product is sometimes known in America as semi-water-gas. Its analysis runs by volume:

Hydrogen,	H	from	18.73	to	26.55
Marsh-gas,	CH_4	"	.31	"	1.11
Olefiant gas,	C_2H_4	"	.31		
Carbonic oxide,	CO	"	25.07	"	18.20
Carbonic acid,	CO_2	"	6.57	"	11.30
Oxygen,	O	"	.03	"	.47
Nitrogen,	N	"	48.98	"	42.28

The ash-pit B is closed and air and steam are forced through N and up through the mass of anthracite or coke which fills the producer-chamber. The feeding is done through the hopper A' by means of its double lid and air-lock action. The gas passes up through the coke-scrubber into the holder K.
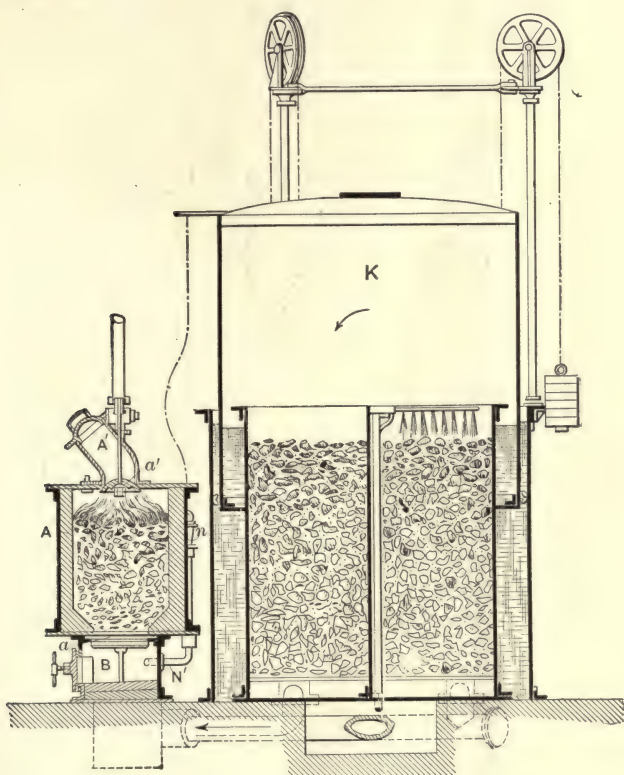
Fig. 11.

A French form of water-gas producer is known as Lencauchez'. Its object is to improve on the Dowson type by saving waste heat, and render it available for coals having some tendency to fuse together from the presence of tarry matters (Fig. 12). The hanging bridge E forces the gases

above the middle of the fuel-bed to pass downwards before escaping to the flue F, and so out to the holder through the
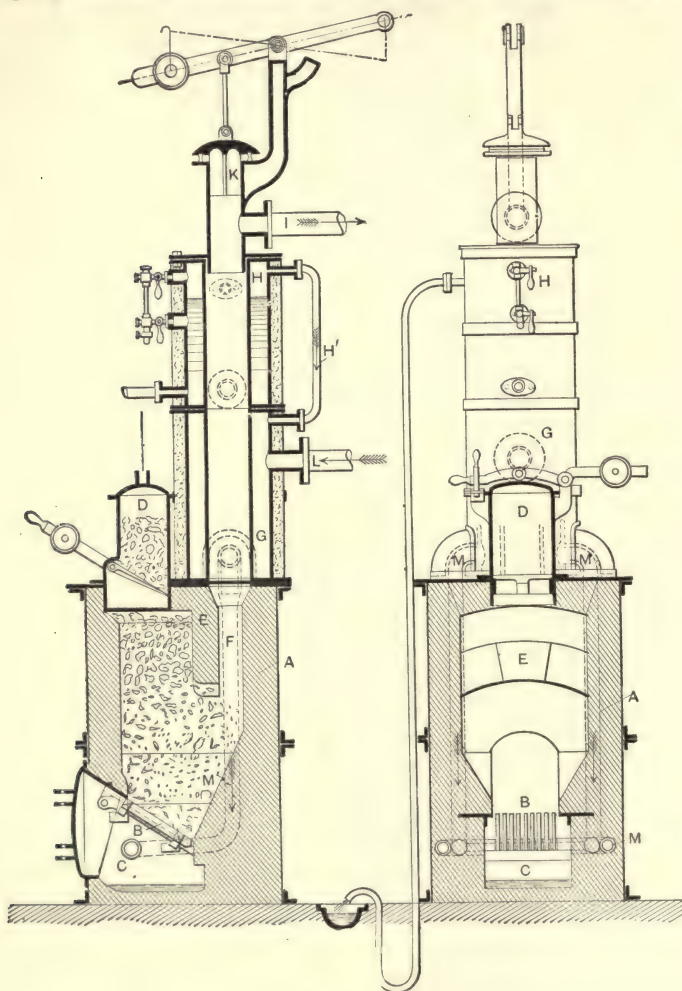
Fig. 12.

passage I. The annular chamber H is a steam-boiler, whose water cools the outflowing gases, and whose steam entering the chamber G meets with air from a blower through the pipe

L, and the combined air and steam are forced through the pipes *M* into the closed ash-pit and so up through the fuel. The descent of the distilled gas through the hot fuel before passing out is the feature which is expected to break up the tarry elements of the distillation. Lencauchez' gas analysis shows:

Hydrogen,	H	18.34
Olefiant gas,	CH_4	1.25
Hydrocarbons,	C_4H_4	1.55
Carbonic oxide,	CO	27.32
Carbonic acid,	CO_2	3.60
Sulphur dioxide,	SO_2	.04
Hydrogen disulphide,	H_2S	.06
Nitrogen,	N	47.84

Dowson gas has a calorific value averaging 150 B.T.U. per cubic foot, while the true water-gas should have 290.

54. Coal-gas or Illuminating-gas.—The ordinary gas used in cities and large towns, and which was universal previous to the introduction of water-gas, is made by distilling bituminous coal in retorts. These retorts are long semi-cylindrical tubes holding each from 160 to 300 pounds of caking bituminous coal—often enriched by some cannel-coal—under and around which the heat from a coke-fire is maintained. The vapors distilled off become a fixed gas by being passed through that part of the distilling apparatus which is kept at a white heat. Other features of the process involve the methods for condensing tarry and offensive vapors and for cleansing, which are aside from the present purpose. The products of distillation of 100 pounds of ordinary gas-coal are usually:

Coke	64	to	65	pounds
Purified gas	15	"	12	"
Ammonia liquor	10	"	12	"
Tar	6.5	"	7.5	"
Loss and impurities	4.5	"	3.5	"
	100.0		100.0	

The composition by volume usually ranges:

Hydrogen.............................. 38 to 48 per cent
Carbonic oxide...................... 2 "   14   "
Marsh-gas, CH₄....................... 43 "   31   "

Heavy hydrocarbons $\left\{\begin{array}{l}\text{Ethylene}\\\text{Propylene}\\\text{Benzole vapor}\end{array}\right\}$ 7.5 " 4.5   "

Nitrogen.............................. 1 "   3   "

The following analyses are taken from a report of Dr. Gideon E. Moore on the Granger Water-gas, 1885:

ANALYSES OF WATER-GAS AND COAL-GAS COMPARED.

	Composition by Volume.			Composition by Weight.		
	Water-gas.		Coal-gas. Heidelberg.	Water-gas.		Coal-gas.
	Worcester	Lake.		Worcester	Lake.	
Nitrogen.............	2.64	3.85	2.15	0.04402	0.06175	0.04559
Carbonic acid.......	0.14	0.30	3.01	0.00365	0.00753	0.09992
Oxygen	0.06	0.01	0.65	0.00114	0.00018	0.01569
Ethylene............	11.29	12.80	2.55	0.18759	0.20454	0.05389
Propylene	0.00	0.00	1.21	....	....	0.03834
Benzole vapor	1.53	2.63	1.33	0.07077	0.11700	0.07825
Carbonic oxide	28.26	23.58	8.88	0.46934	0.37664	0.18758
Marsh-gas..........	18.88	20.95	34.02	0.17928	0.19133	0.41087
Hydrogen	37.20	35.88	46.20	0.04421	0.04103	0.06987
	100.00	100.00	100.00	1.00003	1.00000	1.00000
Density: Theory....	0.5825	0.6057	0.4580	....	....	....
Practice ...	0.5915	0.6018	....	....	....	....
B. T. U. from 1 cu. ft.: water-liquid..	650.1	688.7	642.0	....	....	....
" vapor ..	597.0	646.6	577.0	....	....	....
Flame-temp., Fahr...	5311.2°	5281.1	5202.9°	....	....	....
Av. candle-power...	22.06	26.31	....	....	....	....

CALORIFIC EQUIVALENTS OF CONSTITUENTS OF ILLUMINATING-GAS.

	Heat-units from 1 lb.	
	Water-liquid.	Water-vapor.
Ethylene	21,524.4	20,134.8
Propylene.........................	21,222.0	19,834 2
Benzole vapor.....................	18,954.0	17,847.0
Carbonic oxide....................	4,395 6	4,395.6
Marsh-gas	24,021.0	21,592.8
Hydrogen..........................	61,524.0	51,804.0

55. Acetylene-gas.—The gas C_2H_2 released from calcium carbide by addition of water is as yet of no significance for large-scale heating, but has been much examined for use in motor-carriages and elsewhere where gas-power in small bulk is the prerequisite. One pound of calcium carbide with a half-pound of water will liberate $5\frac{3}{4}$ cubic feet of gas. It has a heat capacity of 18,260 B.T.U. per pound or 1259 per cubic foot, at $14\frac{1}{2}$ cubic feet to the pound. It requires $12\frac{1}{2}$ volumes of air to burn it, which is usually raised to 14 or 15 in practice. It has been compressed to a liquid at 68° F. by a pressure of 600 pounds per square inch.

Acetylene ignites at 510° F. in proper mixtures with air.

56. Comparison of Gaseous Fuels.—Of the four sorts of gas used as a source of heat, water-gas has the highest theoretical temperature of combustion—4850° F. Producer-gas gives 3441°. The natural gas and coal-gas are nearly of the same value as the water-gas. The following tables quoted from Mr. Emerson McMillin give some interesting facts.

Including natural gas the relative volumes and weights of gaseous fuels are:

	By Weight.	By Volume.
Natural gas..............	1000	1000
Coal-gas................	949	666
Water-gas...............	292	292
Producer-gas............	76.5	130

COMPOSITION OF GASES BY VOLUME.

	Findlay, O., Natural Gas.	Coal-gas.	Water-gas.	Penna. Steel-works Producer-gas.
Hydrogen	2.18	46.00	45.00	6.00
Marsh-gas.....................	92.60	40.00	2.00	3.00
Carbonic oxide	0.50	6.00	45.00	23.50
Olefiant gas..................	0.31	4.00	0.00	0.00
Carbonic acid	0.26	0.50	4.00	1.50
Nitrogen	3.61	1.50	2.00	65.00
Oxygen	0.34	0.50	0.50	0.00
Water-vapor	0.00	1.50	1.50	1.00
Sulphydric acid...............	0.20	....	....	....
	100.00	100.00	100.00	100.00

COMPOSITION OF GASES BY WEIGHT.

	Findlay, O., Natural Gas.	Coal-gas.	Water-gas.	Penna. Steel-works Producer-gas.
Hydrogen	0.268	8.21	5.431	0.458
Marsh-gas.....................	90.383	57.20	1.931	1.831
Carbonic oxide................	0.857	15.02	76.041	25.095
Olefiant gas	0.531	10.01	0.000	0.000
Carbonic acid.................	0.700	1.97	10.622	2.517
Nitrogen	6.178	3.75	3.380	69.413
Oxygen........................	0.666	1.43	0.965	0.000
Water-vapor...................	0.000	2.41	1.630	0.686
Sulphydric acid...............	0.417	....	....	....
	100.000	100.00	100.600	100.000

TABLE OF RELATIVE EVAPORATION OF WATER IN A STEAM-BOILER.

	Natural-gas.	Coal-gas.	Water-gas.	Producer-gas.
Cubic feet gas......	1000	1000	1000	1000
Pounds of water....	893	591	262	115

TABLE OF RELATIVE COSTS OF GASES PER MILLION B.T.U. WHICH THEY ARE THEORETICALLY ABLE TO PRODUCE.

Cents per Million B.T.U.

Coal-gas........ 734,976 units, costing 20.00 cents.......... 27.21
Water-gas ...... 322,346 " " 10.88 " 33.75
Producer-gas.... 117,000 " " 2.58 " 22.05

E. P. Reichhelm has discussed the use of gaseous fuel for forge fires, for drop-forging, in annealing-ovens, and furnaces for melting brass and copper, for case-hardening, muffle-furnaces, and kilns. Under ordinary conditions, in such furnaces he estimates that the loss by draft, radiation, and the heating of space not occupied by work is with coal 80 per cent, with petroleum 70 per cent, and with gas above the grade of producer-gas 25 per cent. He gives the following table of comparative cost of fuels, as used in these furnaces:

Kind of Gas.	Number of Heat-units in 1000 cu. ft. used.	Number of Heat-units in Furnaces after deducting 25% Loss.	Average Cost per 1000 ft.	Cost of 1,000,000 Heat-units obtained in Furnaces.
Natural gas....	1,000,000	750,000	... .	... .
Coal-gas, 20 candle-power.................	675,000	506,250	$1.25	$2.46
Carbureted water-gas	646,000	484,500	1.00	2.06
Gasolene gas, 20 candle-power.............	690,000	517,500	.90	1.73
Water-gas from coke	313,000	234,750	.40	1.70
Water-gas from bituminous coal	377,000	282,750	.45	1.59
Water-gas and producer-gas mixed........	185,000	138,750	.20	1.44
Producer-gas	150,000	112,500	.15	1.33
Naphtha-gas, fuel 2½ gallons per 1000 ft...	306,365	229.774	.15	.65
Coal, $4 per ton, per 1,000,000 heat-units utilized.......................				.73
Crude petroleum, 3 cts. per gal., per 1,000,000 heat-units.........................				.73

In the use of natural gas in boiler-firing, it has been found that one man can attend to 1500 horse-power of boilers, while with coal he could handle only 200 horse-power; hence a ratio of 2 to 15 in labor cost would show that where wages are $2.50 per day and the boilers require 4 pounds of coal per horse-power per hour, the cost of firing with coal will be, per ton of coal per 12-hour shift,

$$\frac{2.5 \times 2000}{12 \times 200 \times 4} \text{dollars} = 52 + \text{cents,}$$

while tor gas but $\frac{2}{15}$ of this will be required. Hence the following table will show the relative costs of coal and gas (water) with and without difference in labor cost:

Cost of Coal delivered to Boiler-room per Ton of 2000 lbs.	Price of Gas per 1000 cubic feet.			
	Not including Difference in Cost of Attendance.		Including Difference in Cost of Attendance.	
	Carbureted.	Uncarbureted.	Carbureted.	Uncarbureted.
$6.00	14.8 cts.	6.5 cts.	16.0 cts.	7.1 cts.
5.00	12.3 "	5.4 "	13.6 "	6.0 "
4.00	9.9 "	4.4 "	11.1 "	4.9 "
3.00	7.4 "	3.3 "	8.6 "	3.8 "
2.00	4.9 "	2.2 "	6.2 "	2.7 "
1.00	2.5 "	1.1 "	3.7 "	1.6 "

The uncarbureted gas, exclusive of interest on capital, costs from 10 to 20 cents per 1000 cubic feet. Hence the advantages in the use of gas are from other directions than economy. The calorific power of the uncarbureted gas is about 54 per cent of that of the carbureted gas in the above table.

The heating value of New York City illuminating-gas, as used in gas-engines and for general heating, has been reported by Mr. E. G. Love, per cubic foot at 60° F. and barometer at 30 inches:

715, 692, 725, 732, 691, 738, 735, 703, 734, 730, 731, 727, giving an average of *721*.

Probably 710 heat-units would be more nearly representative of average good quality. The coal-gas of London, with 16 to 17 candle-power, has a calorific power of 668 units per foot and costs from 60 to 70 cents per thousand cubic feet. It ignites at temperatures of 750° to 800° F. with proper mixtures of air.

57. Powdered Fuel.—Among the many experiments which have been tried to utilize waste coal in dust or culm from mine or breaker refuse is to be mentioned the plan of reducing the coal to powder, and blowing it into the furnace in the state of fine division. The pulverizers may be of any

type; the fuel-dust is injected by means of an injector such as is used for oil-vapor systems, using air as the conveying medium. The furnace has to be lined with fire-brick which must be first brought to a high temperature by an open fire. Then the powder is blown in, and once ignited burns regularly and with good economy and without smoke, maintaining an intense heat.

58. Calorific Power of a Hydrocarbon.—It has been already said that the calorific power of a compound was the sum of the calorific power of its components (§ 22). Thus for olefiant gas, C_2H_4, made up of

$$C_2 + H_4 = 24 + 4 = 28 \text{ parts by weight,}$$

$\frac{4}{28} = \frac{1}{7}$ will be hydrogen, and $\frac{24}{28} = \frac{6}{7}$ will be carbon. If then $\frac{1}{7}$ of the calorific power of hydrogen be added to $\frac{6}{7}$ of the calorific power of carbon, their sum will be the calorific power of the compound. With analyzed hydrocarbons the percentages of each constituent will be used instead of the fraction above.

The accepted formula for computing the calorific power from an analysis is due to the physicist Dulong and is known by his name. It is:

$$\text{Calorif. power of 1 lb. in B.T.U.} = 14500C + 62500\left(H - \frac{O}{8}\right).$$

In this C, H, and O are the percentages respectively of carbon, hydrogen, and oxygen, divided by 100 to reduce them to actual fractions of one pound. This is often transformed by the expedient of factoring the constants denoting the respective calorific powers of carbon and hydrogen so as to read:

$$\text{Calorif. power} = 14500\left[C + 4.31\left(H - \frac{O}{8}\right)\right],$$

since $\dfrac{62500}{14500} = 4.31$.

For an analyzed gas, when the weight of a cubic foot is known and the proportion of each combustible in such cubic foot, the calorific power is found by multiplying each weight as a fraction of the whole by the calorific power of a whole unit of that combustible and adding these products together. The calculated calorific powers in the following tables (§ 60) were calculated in this way.

59. Evaporative Power of a Fuel. Efficiency. Heat-balance.—When a fuel is used for making steam to be used as a medium in an engine-cylinder, the weight of water at 212° F. which that fuel will make into steam at atmospheric pressure becomes of interest, and is a standard unit of comparison between fuels. Experiment has shown that to change the state of water at 212° into steam at 212° requires an absorption of 966 British thermal units per pound of water so evaporated. Hence if the calorific power of the fuel be divided by 966, the quotient will be the maximum evaporative capacity of that fuel. For pure carbon the calculation is

$$E = \frac{14500}{966} = 15 \text{ pounds of water from and at } 212°.$$

This figure, of course, is never reached in actual firing of boilers. Eighty per cent of the total heat can be obtained in special conditions; 70 to 75 in regular practice with good anthracite coal. What are the reasons why the theoretical efficiency is not attained?

1. The loss in raising the air and the resulting products of combustion from the temperature of the fire-room to that of the flues and chimney-stack. This loss will be diminished by preheating the air by some waste heat, and by diminishing the weight of diluent air in excess of that needed for combustion.

2. The loss in evaporating any water in the coal, and in

the air itself. This moisture has to be raised to 212° and made into steam at atmospheric pressure.

3. The heat lost in heating up the earthy matter or ash in the coal from the temperature of the fire-room to that of the fire. Obviously, therefore, the less the percentage of such ash the more effective the fuel.

4. The heat absorbed in distilling off from the fuel the volatile matter or hydrocarbons, if any, and raising them and the solid carbon of the fresh-fuel charge to the point at which they will burn. This often makes the softer bituminous coals show to a disadvantage in competition with hard coal containing less volatile matter.

5. The loss of unburned carbon dropping down with the ash and removed with it.

6. Radiation losses from the furnace outwardly to the air. In brick-set furnaces the mass of brick absorbs heat in starting, which it returns in part at the latter part of a run; but the continuous transfer after stationary conditions are reached is never regained. These are largely unavoidable. Preventable losses will arise from:

7. Unnecessarily high stack temperature, due either to excessive firing or to inability of an inadequate absorbing surface in the boiler to take care of heat supplied to it.

8. Incomplete combustion, with loss of available carbon up the chimney-stack as gas or as smoke.

9. Excess of diluting air either below the grate or above it, or behind the bridge-wall.

10. Loss of solid carbon as sparks or cinders with a strong draft.

Many of these are interrelated to each other, and will furthermore be affected by size of the fuel selected for use. Their diminution by mechanical stoking and forced draft conditions will be discussed in subsequent chapters.

A comparison of water evaporated by different qualities of fuel has been made by Mr. Geo. H. Barrus, in which

broken anthracite is assumed to have a capacity of 11 pounds of water if free from ash. The figure given in the table for this fuel is that corresponding to 11 per cent of ash.

Kind of Coal.	Water Evaporated from and at 212° by One Pound of Dry Coal.	Relative Efficiency in Per Cent. Cumberland = 100.
Cumberland	11.04	100
Anthracite, broken...........................	9.79	89
Anthracite, chestnut	9.40	85
Two parts pea and dust and one part Cumberland	9.38	85
Two parts pea and dust and one part culm	9.01	82
Anthracite, pea	8.86	80
Nova Scotia culm	8.42	76

An interesting computation of the results and requirements with a combustion of 100 pounds of anthracite is given in the following table, where it is assumed that the coal and air have a temperature of 60° and that the chimney-gases are at 500°. Hot ashes are withdrawn at 450°, and 2 per cent of carbon goes out with them. Under the assumed conditions 21 per cent is lost, for which ash and moisture in coal and air are responsible for over 5 per cent.

HEAT-LOSSES INCIDENT TO THE COMBUSTION OF 100 POUNDS ANTHRACITE COAL.

Heat-losses.	Number of B. T. U.	Per cent of Total Heat of Fuel.
By water $= [(212-60) \times$ wt.$] + 965.7 \times$ wt. $+ [$sp. heat $\times (500-212) \times$ wt.$]$	37012.5	2.83
By carbonic acid $=$ wt. $\times$ sp. heat $\times (500-60)$..........................	27994.2	2.13
By nitrogen $=$ wt. $\times$ sp. heat $\times (500-60)$	158452.8	12.07
By free oxygen $=$ wt. $\times$ sp. heat $\times (500-60)$............	21973.6	1.67
By ash $=$ wt. $\times$ sp. heat $\times (450-60)$.	1105.7	0.08
By carbon in ash $=$ wt. $\times$ sp. heat $\times (450-60) +$ wt. $\times$ 14650.........	29488.3	2.24
By carbonic oxide $=$ wt. $\times$ sp. heat $\times (500-60) +$ wt. $\times 4400$.........	....	....
Total heat lost exclusive of loss by radiation........................	276027.1	21.02
Theoretically possible evaporation in pounds of water from and at 212° per pound of combustible utilized...		12.73
Theoretically possible evaporation in pounds of water from and at 212° per pound of fuel utilized........		10.44

Pounds.

		Pounds.
100 lbs. of coal.	Water	2.00
	Ash....	11.50
	Carbon	82.00
	Hydrogen	2.00
	Oxygen..........	1.60
	Nitrogen	0.90

Entering furnace.

		Pounds.
1929.83 lbs. of air.	Oxygen for CO_2.	213.33
	Oxygen for H_2O.	14.40
	Oxygen for CO..	00.00
	Oxygen free.	227.73
	Nitrogen	1474.37
	Water............	9.50

WASTE PRODUCTS IN CHIMNEY.

	Pounds.	Per Cent by Wt.
Steam....	29.50	1.46
CO_2......	293.33	14.48
Nitrogen.	1475.27	72.82
CO.......	00.00	00.00
Oxygen..	227.73	11.24

WASTE PRODUCTS IN ASH-PIT.

	Pounds.	Per Cent by Wt.
Ash.....	11.50	85.18
Carbon .	2.00	14.81

TOTAL HEAT OF FUEL.

Weight of C $\times$ 14,650 = 82 $\times$ 14,650 = ... 1,201,300 B. T. U.

Weight of H $- \left(\dfrac{\text{Wt. of O}}{8}\right) \times 62,100 = 2 - \left(\dfrac{1.6}{8}\right) \times 62,100 = 111,780$ "

1,313,080 "

HEAT GENERATED.

80 $\times$ 14,650 = ... 1,172,000 B. T. U.

$2 - \left(\dfrac{1.6}{8}\right) \times 62,100 = ...$ 111,780 "

1,283,780 "

This method is interesting as presenting for the purpose in hand, the practice of striking a heat-balance in any test of a heat appliance. It will be noted that the heat-engine or boiler will be charged with the total quantity of heat energy

delivered to it from all sources during a test, and credited with the units of heat delivered from it in all directions as observed in the test or assumed from experience with like apparatus.   The form of balance-sheet would be as follows:

<div align="center">HEAT-BALANCE.</div>

Dr.   *Cr.*
To heat   By heat
  from coal,   in dry steam,
  from air,   in moisture and water mechanically suspended in
  from feed-water.   steam,
    in dry flue-gases,
    in moisture in coal,   ⎞ at temper-
    in water resulting from combustion,   ⎬ ature of
    in vapor in air,   ⎠ flue-gases.
    lost through incomplete combustion to CO,
    in ashes,
    lost by radiation and otherwise unaccounted for.

Of two appliances for utilizing heat energy, that is the more effective which most completely renders the available heat into useful work or product.   Examples of the distribution of the available heat as reported by various authorities are given in the following table:

<div align="center">DISPOSITION OF HEAT IN STEAM-BOILERS.</div>

Disposition of Heat.	Bunté.	Scheurer and Meunier.		Donkin and Kennedy.			Hoadley.
		A	B	C	D	E	
Waste in flue-gases, including evaporation of moisture in coal and heating vapor in air when these losses are not separately given..	18.6	5.5	14.8	9.4	22.5	6.5	5.04
Evaporating moisture in coal...	3.5	2.5	6.1	0.1	0.1	0.0	1.55
Heating vapor in air.........................	....	.....	......	......	......	....	0.18
Imperfect combustion...........	8.0	6.0	.. ...	12.7	0.0	0.0	1.44
Clinker and ash.......	4.1	......	.....	0.1	0.2	0.0	......
Radiation and heat not otherwise accounted for...............	7.6	23.5	13.4	13.9	11.0	15.0	4.00
Heating and evaporation of water..	58.2	61.0	65.7	63.8	66.2	78.5	87.79

60. Data Concerning Fuels.—In the following tables gathered from various sources are grouped summaries of the data which have been discussed in the foregoing paragraphs. The results are from experiment and analysis.

HEAT OF COMBUSTION OF FUELS.

Kind of Coal.	Analysis.					Per cent of Volatile Matter, exclusive of Water and Ash.	Calorific Power observed.	Calorific Power, excluding Water and Ash.	Calorific Power calculated.
	Carbon.	Hydrogen.	Oxygen and Nitro-gen.	Hydro-scopic Water.	Ash.				
Anthracite from Pennsylvania	85.456	1.995	2.199	3.450	5.900	3.00	7484	8256	8450
Anthracite from Mure	86.564	1.367	2.969	4.700	4.400	2.75	7468	8216	8172
Semi-anthracite from Kébao	85.746	2.733	2.731	2.800	4.450	5.20	7828	8532	8531
Semi-anthracite from Commentry	84.928	2.892	5.005	1.775	5.400	3.19	7850	8456	8360
Semi-anthracite from Blanzy	82.746	2.916	6.278	1.760	6.300	6.00	7773	8203	8215
Semi-bituminous from Anzin	88.473	4.139	4.338	1.350	1.700	14.08	8392	8656	8767
Semi-bituminous from Aniche	85.937	4.198	5.240	0.625	4.000	11.93	8426	8834	8688
Bituminous from Anzin	83.754	4.385	5.761	1.100	5.000	21.51	8051	8574	8689
Bituminous from the collieries of Saint-Etienne	84.546	4.772	5.432	1.250	4.000	20.84	8392	8857	8829
Gas-coal from Béthune	82.418	5.089	7.193	1.200	4.100	30.41	8210	8668	8710
Gas-coal from Commentry	80.182	5.245	8.173	3.000	3.400	39.96	7870	8408	8644
Wigan cannel-coal	78.382	5.060	5.058	0.600	10.900	31.64	7761	8768	9011
Lignite from Styria	65.455	4.782	24.303	0.710	4.750	50.34	6284	6646	6610
Lignite from Vaugirard	59.795	4.512	25.799	3.144	6.750	49.95	5536	6076	6270
Norwegian pine, partially dried	47.366	5.581	39.780	6.940	0.333	68.93	4477	4828	4947
Cellulose, $C_{12}H_{10}O_{10}$	44.440	6.170	49.390	6.920	0.750	68.93	4200	4200	4264
Tourbe from Bohemia	53.183	5.542	34.230	6.125	0.920	68.93	5489	5903	5609
Coke, Commentry coal	92.727	0.444	2.629	0.500	4.200	68.93	7665	8001	7957
Coke, semi-bituminous coal	94.582	0.633	1.585	0.500	3.200	68.93	7787	8044	8130
Coke, Pennsylvania anthracite	91.036	0.685	2.146	0.233	5.900	68.93	7528	8036	8078

TABLE OF AMERICAN COALS.

COAL. Name or Locality.	Constituents in Per Cent of Total Weight.					Fuel Value per Lb. of Coal.		
	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	B. T. U. Calculated.	B. T. U. by Calorimeter.	Theoretical Evaporation in Lbs. from and at 212°.
ARKANSAS.								
Coal Hill, Johnson Co.................	1.35	14.93	74.06	9.66	3.04	13713	....	14.10
Coal Hill, Johnson Co.................	1.70	14.60	74.91	8.79	3.04	... .	11812	12.22
Huntington Co.	1.30	18.95	71.51	8.24	0.78	......	11756	12.17
Huntington Co......................	1.30	18.90	73.15	6.65	0.75	......	11907	12.32
Huntington Co......................	1.27	18.89	71.74	8.10	0.65	......	12537	12.97
Lignite	.....	.....	.....	.....	......	9215	......	9.54
Jenny Lind, Sebastian Co........... ..	1.26	17.64	72.48	8.62	2.11	13964	......	14.40
Spadra, Johnson Co.................	1.47	13.27	78.63	6.63	1.60	14420	......	14.90
COLORADO.								
Lignite........	.....	.....	.....	.....	.. ..	13560	......	14.04
Lignite...........................	.....	.....	.....	.....	......	13865	......	14.35
Lignite, slack	14.80	32.00	42.86	10.34	0.76	8500	......	8.80
Lignite, slack, North Colorado.........	18.88	31.74	40.08	9.30	0.61	......	.....	....
Rouse Mine......................	3.13	37.32	30.00	8.25	... ..	.....	....	....
ILLINOIS.								
Big Muddy, Jackson Co....	7.30	28.28	53.87	10.46	0.98	.....	11466	11.87
Big Muddy, Jackson Co.	6.12	30.95	53.74	9.19	1.22	......	11520	11.93
Big Muddy, Jackson Co.	5.85	31.84	55.72	6.59	2.92	......	11781	12.19
Big Muddy, Jackson Co..............	6.35	31.50	55.25	6.90	2.02	12567	......	13.00
Bureau Co............	.....	.....	.....	.....	.. ..	13025	......	13.48
Colchester	11.60	25.02	44.76	18.62	......	......	9848	10.19
Colchester Slack	5.30	25.45	38.15	31.10	1.20	......	9035	9.35
Collinsville, Madison Co....	9.20	45.89	31.57	13.34	5.34	......	10143	10.50
Dumferline Slack.................	9.64	28.86	39.48	22.02	....	......	9401	9.73
Duquoin Jupiter, Perry Co	11.30	30.31	49.91	8.48	0.91	......	10710	11.08
Ellsworth, Macoupin Co..........	9.26	42.22	42.17	6.35	2.62	12175	.. ...	12.60
Gillespie, Macoupin Co....	12.61	30.58	45.27	11.54	1.45	......	9739	10 09
Girard, Macoupin Co...............	9.70	34.39	45.76	10.15	3.49	......	9954	10.30
Girard, Macoupin Co..............	8.90	32.25	42.89	15.96	8.10	......	10260	10.63
Heitz Bluff, St. Clair Co...........	8.95	37.81	48.24	5.00	3.27	... ..	10332	10.69
Johnson's, St. Clair Co....	5.50	40.14	40.53	13.83	4.80	11723	....	12.10
Loose's, Sangamon Co......	10.71	37.62	45.07	6.60	2.39	11479	......	11.90
Mercer Co	.....	.....	.....	.....	......	13123	......	13.58
Montauk Co...........	....	....	....	....	...	12659	....	13.10
Mt. Olive, Macoupin Co..............	10.38	36.68	46.10	6.84	3.53	11763	......	12.20
Oakland, St. Clair Co..............	8.30	34.40	43.12	14.18	4.42	......	10395	10.76
Reinecke, St. Clair Co.	7.56	39.81	42.49	10.14	4.02	11720	....	12.10
Riverton, Sangamon Co.... ...	11.06	37.94	42.98	8.02	3.27	11406	....	11.80
St. Clair........................	7.80	30.69	39.68	21.83	9.62	......	9261	9.58
St. Clair...	10.25	33.10	41.79	14.86	6.92	......	10294	10 65
St. Clair..............	11.15	34.19	44.94	9.72	4.27	.. ..	10647	11.02
St. Bernard....	14 36	30.86	48.39	6.39	1.38	......	10080	10.44
St. John, Perry Co..................	9.82	28.35	45.77	16.08	2.06	......	9765	10.10
St. John, Perry Co	13.60	34.46	43.54	15.40	1.83	......	9828	10.18
Streator, LaSalle Co..............	12 01	35.32	48.78	3.89	2.38	......	11403	11.80
Trenton, Clinton......................	13.34	30.39	51.96	4.31	0.92	....	10584	10.96
Trenton, Clinton	9.95	31.04	52.03	6.98	1.04	......	11245	11.63
Vulcan nut, St. Clair Co	7.44	30.86	45.09	16.61	1.32	......	9450	9.78
Vulcan nut, St. Clair Co.	10.30	27.91	48.99	12.80	0.71	......	10626	11.00
INDIANA.								
Block......	3.50	32.50	63.10	1.00	0.98	14020	......	14.50
Block..............................	.....	.....	.....	.....	......	13588	......	14 38
Caking............	.....	.....	.. ..	.....	......	14146	......	14.64
Cannel...........................	...	.....	.....	.....	......	13097	......	13 56

TABLE OF AMERICAN COALS.

COAL. Name or Locality.	Constituents in Per Cent of Total Weight.					Fuel Value per Lb. of Coal.		
	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	B. T. U. Calculated.	B. T. U. by Calorimeter.	Theoretical Evaporation in Lbs., from and at 212°.
INDIAN TERRITORY.								
Atoka	6.66	35.42	51.32	6.60	3.73	......	11088	11.47
Choctaw Nation	1.59	23.31	66.85	8.25	1.18	......	12789	13.23
IOWA.								
Good Cheer	10.85	30.32	31.38	27.45	7.32	......	8702	9.01
KENTUCKY.								
Caking	......	......	......	......	......	14391	.....	14.89
Cannel	......	......	......	......	......	15198	.....	16.76
Cannel	......	......	......	......	......	13360	.....	13.84
Lignite	......	......	......	......	......	9326	.....	9.65
MISSOURI.								
Bevier Mines	......	......	......	......	......	......	9890	10.24
MARYLAND.								
Cumberland	......	......	......	......	......	12226	.....	12.65
George's Creek	.....	......	......	......	......	13500	.....	13.98
NEW MEXICO.								
Coal	2.35	35.53	50.24	11.88	0.61	......	11756	12.17
OHIO.								
Briar Hill, Mahoning Co	2.47	31.83	64.25	1.45	0.56	13714	......	14.20
Hocking Valley	8.25	35.88	53.15	2.72	0.43	13414	......	13.90
PENNSYLVANIA.								
Anthracite	......	.....	......	......	.....	14199	......	14.70
Anthracite	......	......	.....	......	.....	13535	......	14.01
Anthracite	.....	.....	.....	......	.....	14221	......	14.72
Anthracite, pea	2.04	6.36	78.41	13.19	.....	12300	.....	12.73
Anthracite, buckwheat	3.88	3.84	81.32	10.96	0.67	12200	......	12.63
Cannel	.....	.....	......	.....	.....	13143	......	13.60
Connellsville	.....	.....	......	.....	.....	13368	... ..	13.84
Pittsburgh (av.)	1.80	35.34	54.94	7.92	1.97	......	13104	13.46
Pittsburgh (coking)	1.43	30.22	61.87	6.48	1.35	14415	.	14.90
Youghiogheny	1.96	34.06	58.98	5.00	.....	......	12936	13.39
Youghiogheny	2.02	32.14	58.66	6.88	0.88	......	12600	13.03
Reynoldsville	1.20	27.12	65.88	5.80	.....	......	12981	13.44
TENNESSEE.								
Glen Mary, Scott Co	2.15	31.47	61.63	4.75	0.94	......	13167	13.63
TEXAS.								
Ft. Worth	14.42	30 03	42.53	13.02	1.47	......	9450	9.78
Ft. Worth	4.60	34.72	49.27	11.41	1.56	...	11403	11.80
Lignite	.....	.. ..	.....	.....	.....	12962	.....	13.41
WEST VIRGINIA.								
New River	......	......	....	....	.....	14200	.....	14.70
New River	......	.. ..	....	....	.....	13400	.....	13.87
New River	0.94	18.19	75.89	4.68	0.30	. ...	.....	... .
New River	0.76	18.65	79.26	1.11	0.23	.. ..	.....	.. ....

COMPOSITION OF COALS, PER CENTS.

Description of Coal.	Carbon.	Hydrogen.	Oxygen.	Nitrogen.	Sulphur.	Ash.	Specific Gravity.
ANTHRACITES:							
French	90.9	1.47	1.53	1.00	0.80	4.30	1.35
Welsh	91.7	3.78	1.30	1.00	0.72	1.50	1.37
Rhode Island	85.0	3.71	2.39	1.00	0.90	7.00	1.42
Pennsylvanian	78.6	2.50	1.70	0.80	0.40	14.80	1.45
SEMI-BITUMINOUS:							
Maryland	80.0	5.00	2.70	1.10	1.20	8.30	1.33
Welsh	88.3	4.70	0.60	1.40	1.80	3.20	.... ..
BITUMINOUS:							
Pennsylvanian	75.5	4.93	12.35	1.12	1.10	5.00	1.32
Indiana	69.7	5.10	19.17	1.23	1.30	3.50	1.25
Illinois	61.4	4.87	35.42	1.41	1.20	5.70	1.31
Virginian	57.0	4.96	26.44	1.70	1.50	8.40	1.32
Alabama	53.2	4.81	32.37	1.62	1.30	6.70	1.30
Kentucky	49.1	4.95	41.13	1.70	1.40	7.20	1.28
Cape Breton	67.2	4.26	20.16	1.07	1.21	6.10	1.33
Vancouver's Island	66.9	5.32	8.76	1.02	2.20	15.80	1.28
Lancashire gas-coal	80.1	5.50	8.10	2.10	1.50	2.70	......
Boghead cannel	63.1	8.90	7.00	0.20	1.00	19.80	....
LIGNITES:							
Californian brown coal	49.7	3.78	30.19	1.00	1.53	13.80	1.32
Australian brown coal	73.2	4.71	12.35	1.11	0.63	8.00	1.27
PETROLEUMS:							
Pennsylvanian, crude	84.9	13.70	1.40	..... .	....	....	0.886
Caucasian, light	86.3	13.60	0.10	....	....	....	0.884
" heavy	86.6	12.30	1.10	....	....	....	0.938
Petroleum refuse	87.1	11.70	1.20	....	....	....	0.938

TABLE SHOWING THE COMPOSITION AND CALORIFIC POWER OF VARIOUS COMBUSTIBLES—THE QUANTITY OF OXYGEN AND AIR NECESSARY FOR COMBUSTION—AND THE VOLUME OF THE PRODUCTS OF COMBUSTION OF I LB. OF COMBUSTIBLE.

Name of Combustible.	Composition.				Calorific Power.	Weight of Oxygen Necessary for Combustion.	Weight of Air Necessary for Combustion.	Vol. of Air Corresponding in Cubic Feet.	Vol. of Products in Cubic Feet.
	C	H	Volatile Matter.	Ashes.					
Carbon	1.00	.....	.....	.....	14400	2.66	11.29	137.6	137.6
Anthracite coal	0.90	0.03	0.03	0.04	13500	2.64	11.21	138.9	136.2
Bituminous coal	0.85	0.05	0.06	0.06	14400	2.66	11.29	139.6	140.1
Lignite	0.70	0.05	0.20	0.05	11700	2.26	9.69	120.2	116.3
Peat	0.55	0.05	0.30	0.10	9000	1.86	7.90	97.9	102.1
Peat, o.20 water	0.39	0.04	0.50	0.07	7200	1.49	6.32	78.3	81.5
Coke	0.85	0.05	.....	0.10	12600	2.26	9.69	120.2	116 9
Peat-charcoal	0.82	.....	.. ..	0.18	9000	2.18	9.25	114.5	112.7
Dry wood	0.48	0.06	0.05	0.01	7200	1.75	7.43	91.9	89.2
Wood, o.20 water	0.40	0.05	0.25	0.01	5400	1.40	5.94	73.5	71.8
Wood-charcoal	0.80	... ..	0.04	0.07	10800	1.86	7.90	97.9	96.3
Hydrogen	.....	1.00	.....	.....	62000	8.00	33.97	420.6	475.4
Carbonic oxide	0.43	.....	0.57	.....	4320	0.57	2.42	20.9	35.6
Illuminating-gas	0.62	0.21	0.17	.....	18000	2.64	11 22	136.3	176.7
Gas from blast-furnace	0.06	0.02	0.92	... ..	1620	0.23	0 99	12.2	30.3

COMPARATIVE COMPOSITION OF GAS.

	Natural Gas.	Coal-gas.	Water-gas.	Producer-gas.	
				Anthra.	Bitu.
CO	0.50	6.0	45.0	27.0	27.0
H	2.18	46.0	45.0	12.0	12.0
CH_4	92.6	40.0	2.0	1.2	2.5
C_2H_4	0.31	4.0	... .	......	0.4
CO_2	0.26	0.5	4.0	2.5	2.5
N	3.61	1.5	2.0	57.0	56.2
O	0.34	0.5	0.5	0.3	0.3
Vapor........	....	1.5	1.5	......	......
Pounds in 1000 cubic feet......	%45.6	32.0	45.6	65.6	65.9
Heat units in 1000 cubic feet ..	1,100,000	735,000	322,000	137,455	156,917

NATURAL GAS IN OHIO, INDIANA, AND PENNSYLVANIA.

Description.	Ohio.			Indiana.				Penn.
	Fostoria.	Findlay	St. Mary's.	Muncie	Anderson.	Kokomo.	Marion.	Pittsburgh.
Hydrogen..........	1.89	1.64	1.94	2.35	1.86	1.42	1.20	22.
Marsh-gas..........	92.84	93.35	93.85	92.67	93.07	94.16	93.57	67.
Olefiant gas........	.20	.35	.20	.25	.47	.30	.15	1.
Carbon monoxide ..	.55	.41	.44	.45	.73	.55	.60	0.
Carbon dioxide.....	.20	.25	.23	.25	.26	.29	.30	0.6
Oxygen............	.35	.39	.35	.35	.42	.30	.55	0.8
Nitrogen...........	3.82	3.41	2.98	3.53	3.02	2.80	3.42	3.0
Hydrogen sulphide.	.15	.20	.21	.15	.15	.18	.20	......

Approximately 30,000 cubic feet of gas have the heating power of one ton of coal.

PRODUCER-GAS FROM ONE TON OF COAL.

Analysis by Vol.	Per Cent.	Cubic Feet.	Lbs.	Equal to—
Co	25.3	33,213.84	2451.20	1050.51 lbs. C + 1400.7 lbs. O.
H	9.2	12,077.76	63.56	63.56 " H.
CH_4.............	3.1	4,069.68	174.66	174.66 " CH_4.
C_2H_4............	0.8	1,050.24	77.78	77.78 " C_2H_4.
CO.............	3.4	4,463.52	519.02	141.54 " C + 377.44 lbs. O.
N (by difference)	58.2	76,404.96	5659.63	7350.17 " Air.
	100.0	131,280.00	8945.85	

RELATIVE CALORIFIC VALUES.

	By Weight.	By Volume.
Natural gas...............	1,000	1,000
Coal-gas.................	949	666
Water-gas	292	292
Producer-gas.	76.5	130

The following table gives the accepted power capacity of various sources of heat energy in their relation to motors using these combustibles for power in gas or other engines.

Material.	Heat-units per Pound.	Heat-units per Cubic Foot.
Hydrogen....................................	61560	293.5
Carbon.....	14546	....
Crude petroleum, W. Va., sp. gr. .873..........	18324	....
Light petroleum, Pa., sp. gr. .841..............	18401	....
Benzine C_6H_6..................................	18448	....
Gasoline....................................	11000	....
Illuminating-gas, 28 c. p.......................	....	950
" " 19 "	....	800
" " 15 "	....	620
Water-gas, American.........................	....	185
Producer-gas, English	....	150
Water producer-gas...........................	....	104
Ethylene-olefiant gas C_2H_4.....................	21430	1677
Gasoline vapor...............................	11000	690
Acetylene, C_2H_2...............................	21492	868
Natural Gas, Leechburg, Pa....................	....	584
" " Pittsburg, Pa....................	....	495
Marsh-gas, methane, CH_4......................	23594	1051

CHAPTER VI.

TEMPERATURES OF COMBUSTION. PYROMETERS.

61. Introductory.—It will have been noticed that the calorific power of a fuel, or the total heat liberated from it upon combustion, is a quantity which is independent of the time required for that combustion. It is equally a fact of common observation that when a fuel is forced to burn rapidly the fire is hotter, or the flame has a higher temperature, than when the fuel is burned more gently, and the combustion process is extended over a longer time. In other words, the temperature of the fire is not independent of the time taken for combustion, but varies inversely as such time, while the calorific power is independent of time. This makes it a vital matter that the engineer should be able to control the rapidity of the combustion of the fuel, and this is most easily done by controlling the weight and velocity of the supply of air.

The higher temperature in the fire under rapid or forced combustion is due not only to the fact that under the assumption of a constant condition of intensity more fuel units will be supplied by the fireman or the firing machinery per unit of time when the combustion is more rapid. It is also true that the more intense the chemical activity, the higher the heat which accompanies such activity; and besides, the transfer of heat from the fire to absorbing bodies whereby its temperature is lowered will be greater the longer any given weight of fuel is permitted to occupy the grate or combustion area. This would imply that where transfer is a principal

feature of a combustion—as in a boiler—there may be an economical maximum rate of combustion which it would not be desirable to exceed. And finally, when rapid chemical combination with air is secured, as with oil- or gas-firing, and proper provision is made for admitting air and compelling chemical action by attention to details, a less excess (or none) of free oxygen is required in the products of combustion, and the cooling and diluting effect of the air and its inert nitrogen is diminished. Two questions are then suggested: 1. What is the probable temperature of a fire and how may it be observed? 2. By what means shall the rate of combustion be secured and controlled?

62. Temperature of the Fire.—It will appear to be an obvious deduction from Chapter III (§§ 11 to 14) that when one body conveys or transfers heat to another substance which absorbs it entirely in raising its temperature, the quantity of heat in heat-units so transferred will be represented by the product of the weight (w) by its specific heat (c) by the change in temperature indicated by the difference between its final temperature and its initial temperature ($t_2 - t_1$) if the body in question was hotter after the transfer than it was before. This holds true also for the body or substance which has cooled, since the one has lost heat to the same extent as the other has gained it. In symbols, if Q represents the quantity of heat in units transferred from one body to the other,

$$Q = w \times c \times (t_2 - t)$$

for the body which has become warmed, and

$$Q' = w' \times c' \times (t_2 - t_1)$$

for the body which has cooled, having a different weight and different specific heat. But $Q = Q'$. It has therefore been a convention to assume that the temperature of the fire was the same as that of the flaming products of the combustion

at the instant when combustion was complete; and that all the heat liberated from the fuel was used to raise the products of combustion to this fire temperature. All that is unknown then in the above formula is the final high temperature of the products of combustion, when the total heat transferred, the weight of the products of combustion (§§ 24 to 28) are known, and their specific heat. This theory is sound and applicable for the conditions which prevail in a determination with a coal-calorimeter (§ 23), but the considerations advanced in the preceding paragraph throw some doubt upon its reliability as applied to practice. The assumption is further implied that specific heats are constant at all temperatures. The specific heats under constant pressure of the constituents of the fire are:

For carbonic acid gas.................... 0.217
    "   steam or water-vapor 0.480
    "   nitrogen 0.244
    "   air.............................. 0.238
    "   ashes (probably)................. 0.200

The data for the computation of a hypothetical fire-temperature for two combustibles, such as pure carbon and olefiant gas, will be:

	Carbon, C	Olefiant Gas, C_2H_4
(1) Calorific power..............................	14,500	21,300
(2) Pounds weight of products of combustion......	13.00	16.43
(3) Mean specific heat...........................	0.237	0.257
(4) Specific heat and weight.....................	3.08	4.22
Value for $(t_2 - t_1)$, or (1) ÷ (4).............	4580°	5050°

This assumes no dilution of the products by air. The use of diluting air will act to lower the hypothetical temperature by increasing the weight to be heated. The specific heat of course approaches more nearly to that of air the greater the proportion of air in the products of combustion.

Recent investigations with the pyrometer show that tem-

peratures hitherto have been often overestimated, and that the high values obtained by the above method are not reached in practice. A generally accepted series of data is given in the table below, constructed by M. Pouillet. Its use of course must be subject to the peculiarities of the individual and the degree of general illumination prevalent.

Heat	Deg. C.	Deg. F.	Heat.	Deg. C.	Deg. F.
Incipient red	535	977	Clear orange...........	1200	2192
Dull red...............	700	1292	White heat.............	1300	2372
Incipient cherry	800	1472	Bright white heat.....	1400	2552
Cherry-red	900	1652		1500	2732
Clear cherry-red......	1000	1832	Dazzling white heat. {	to	to
Deep orange	1100	2021		1600	2912

The melting or heating of metals with fuel implies that the heat of the fire should not be less than that demanded by the metal. Recent determinations by Prof. Roberts-Austen give melting-temperatures as follows:

Copper...................... 1929° to 1996° F.
Cast-iron, white.................... 2075
    "   gray..................... 2228
Steel, hard........................ 2570
    "   mild........................ 2687
Wrought iron............... 2732 to 2912
Platinum........................... 3227

The melting-temperature of steel at the end of its conversion in the open-hearth steel process is among the highest usually met in industry, and is about 2732° F. Even this, however, is only attained by preheating both gas and air used in the process. The heat in a furnace for baking hard porcelain may rise to 2500° F. Furthermore, it is a question somewhat of quantity of heat rather than its intensity. The platinum value is that for material which is only fusible before the oxyhydrogen blowpipe. All these data tend to throw a certain doubt upon the validity of the accepted

method of working out flame or fire temperatures, and leave the field open for practical determinations by pyrometer.

Hoadley gives the temperature in a boiler-fire as ranging from 2493° F. to 2793° F. in its heart, and 1340° to 1600° F. at the bridge-wall. In a marine boiler, Durston records 1644° in the combustion-chamber. Clark gives 1705° for a combustion rate of 20 pounds per square foot grate, rising to 2100° at a 120-pound rate.

63. Pyrometers. General.—The usual appliances for measuring temperature (§§ 13 to 15) are not suitable for high heats, with the possible exception of the air-thermometer. Mercury-thermometers with nitrogen gas in the tube above the mercury can be used up to 800° F.; the ordinary construction being open to the objection that the rate of expansion of mercury increases with the rise of temperature, so that a mercury-thermometer which agreed with an air-thermometer at 212° would read low at temperatures below this point, and high at temperatures above it.

The name pyrometer is given to an appliance for measuring or observing high temperatures. There are several principles which have been applied in such instruments.

1. Melting-points of various metals or alloys. This is approximate only, since the melting-point varies with chemical purity of the metals in question, and undergoes change with time, with frequency of melting, and deterioration by heat.

2. Expansion of metals by heat. These may be single bars, or compound bars of two or more metals whose rate of expansion is not the same. Copper and iron are two usual metals to use (as in Brown's pyrometer and Bulkley's); the bar flexes, and the amount of flexure indicates the heat on a convenient dial. To this class belongs the use of clay bars, which contract under heat, as in the Wedgwood pyrometer formerly used by potters for their ovens. Clay, however, is not uniform nor permanent, and in the case of the metal bars

a permanent change occurs by and by in the molecular struc-
ture consequent upon the prolonged heating, which prevents
accuracy.

3. Changes in volume and action of a permanent gas such
as air, either using its expansion, as in the air-thermometer
and nitrogen-thermometer, and in the Wiborgh pyrometer, or
depending on the changes in rate of flow with temperature,
as in the Uehling-Steinbart pyrometer.

4. Methods based upon the specific heat of solids, raised
to the fire-temperature and cooled in water.

5. Time required to heat a weighed quantity of water
enclosed in a vessel, as in the water-pyrometer.

6. Changes in the electric resistance of a refractory con-
ductor such as platinum exposed to heat, as in the Siemens
pyrometer.

7. Measurement of the strength of a thermo-electric cur-
rent produced by heating the junction of two metals, as in
the Le Chatelier pyrometer.

8. Optical effects in deflection of light rays from incan-
descent solids, as in Mesuré and Nouel's pyrometric tele-
scope.

64. Metal-ball Pyrometer.—The metal-ball pyrometer is
one of the easiest to arrange for. A ball or other mass of
metal of known weight and specific heat is put into the place
whose heat is desired, and left there until it gains the same
temperature as the medium around it. It is then withdrawn
and dropped into a known weight of water of observed tem-
perature, and the rise of temperature is observed which it
causes in the water. The principles of transfer demand that
if W is the weight of water with a specific heat of unity and
an initial temperature of $t°$ F., while w is the weight of the
ball, c its specific heat, T the final temperature of the water,
then the unknown temperature x of the ball will be given
from the equality

$$W \times 1 \times (T - t) = w \times c \times (x - T);$$

whence

$$x = \frac{W(T - t)}{wc} + T.$$

For greater accuracy, corrections must be made for variations of specific heat, for the capacity of the cooling vessel itself for heat, for losses in transit from fire to cooling vessel, etc. The ball may be platinum, iron, or other metal, or fire-clay or fire-brick.

65. Wiborgh Air-pyrometer.—This form of pyrometer involves the use of a porcelain globe or cylinder connected through a capillary tube with the exterior air. The air in the globe is heated by the temperature to be measured, with the tube open. Then a known volume of air of known temperature is forced into the globe with its outlet closed, and the resulting pressure observed by a delicate pressure recording-device. Then, since the initial volume and pressure are known before the addition of the cooler air, and the final pressure is observed when the volume and temperature of the added air are known, the only unknown factor is the original temperature, which is the quantity desired.

This form of apparatus is useful for temperatures between 0° and 2400° F., such as occur at metallurgical and similar furnaces.

66. Uehling-Steinbart Pyrometer.—The Uehling pyrometer depends on a principle of the flow of a permanent gas, such as air through a minute aperture, which makes the weight which flows in a given time a function of the density of the air, which varies directly as the absolute temperature. This is applied by having a closed tube or chamber fitted with minute inlet and outlet orifices and causing air to flow in through one and out through the other, by means of proper aspirating appliances, while the tension in the chamber is carefully measured by a sensitive manometer. The air to enter is made to have the desired temperature by locating the inlet aperture in the end of a platinum tube in

the bulb of a porcelain tube over which the hot gases pass, or which is inserted in the chamber or place whose temperature is to be ascertained. The outlet aperture is kept at a lower but constant temperature, and is operated at a constant suction effort by surrounding it with boiling water in a proper coupling and aspirating by means of a water-column kept at a constant height.

67. **Le Chatelier Thermo-electric Pyrometer. Siemens Pyrometer.**—The principle of the thermo-electric pile is used in this apparatus. Two wires, one of platinum and the other of platinum with 10 per cent of rhodium, are made into the usual thermo-electric couple of the physical laboratory, and are presented to the heat to be measured. The action of heat creates a current of electricity in the couple, whose intensity is measured by a galvanometer. The instrument should be calibrated experimentally by heating the junction of the pile to temperatures which are known as observed by the air-thermometer, and plot the curve of deflections from these data. The error of a Le Chatelier pyrometer is usually less than 50°, up to its limit of use. The Siemens pyrometer depends upon the principle that the conductivity of platinum wire is diminished by heat, and measurably to a degree proportional to that temperature if the wire is thoroughly homogeneous. If, therefore, an electric current is divided, and by a Wheatstone bridge or other controllable resistance the two branches are made to have equal resistance at the same known temperature, it becomes easy, by raising one branch of the wire to known temperatures and equating the varying resistance, to make a calibration whereby unknown temperatures may be determined, either by the galvanometer-reading directly, or by the resistance necessary to introduce to keep its reading constant. An uncertainty is introduced, however, by the difficulty of avoiding a molecular change in the conductor which is exposed to heat, of whose extent and effect there is always some uncertainty.

68. Mesuré and Noel's Pyrometric Telescope.—For observing temperature of incandescent or glowing bodies by the color of the rays of light which they emit, an appliance can be easily constructed which shall serve as a pyrometer, basing it on the principle that a plate of quartz cut at right angles to the crystalline axis rotates the plane of polarization of polarized light to a degree nearly inversely proportional to the square of the wave-length of such light. If, then, two Nicols prisms be placed in a tube and a ray of monochromatic light be passed through the first prism or polarizer, and watched through the second or analyzer, with the plate of quartz between them, a part of the light which passed the first and in the absence of the quartz was extinguished will be made visible when the quartz is present. To extinguish it the analyzer must be rotated, and rotated further as the light from the hot body gathers the shorter waves of orange and yellow and emits them all as it passes to the dazzling glow of white light. The degree to which the analyzer must be turned is a measure of the temperature of the radiating body.

Like the foregoing instruments, this should be calibrated experimentally, and has the great advantages of cheapness, convenience, and portability to offset the absence of exact definition of temperature, and the difficulty in applying it to widely varying materials.

69. Some Standard Temperatures.—For use in standardization of other appliances, or for direct use in pyrometry, certain accepted temperatures gathered from various sources are here presented in tabular form.

Deg. F.	Deg. C.		Deg. F.	Deg. C.	
212	100	Water boils.	1733	945	Silver melts.
618	326	Lead melts.	1859	1015	Potassium sulphate melts.
676	358	Mercury boils.			
779	415	Zinc melts.	1913	1045	Gold melts.
838	448	Sulphur boils.	1929	1054	Copper melts.
1157	625	Aluminum melts.	2732	1500	Palladium melts.
1229	665	Selenium boils.	3227	1775	Platinum melts.

BOILING-POINTS AT ATMOSPHERIC PRESSURE.

14.7 lbs. per square inch.

Ether, sulphuric	100° F.	Average sea-water	213.2° F.
Carbon bisulphide	118	Saturated brine	226
Ammonia	140	Nitric acid	248
Chloroform	140	Oil of turpentine	315
Bromine	145	Phosphorus	554
Wood-spirit	150	Sulphur	570
Alcohol	173	Sulphuric acid	590
Benzine	176	Linseed oil	597
Water	212	Mercury	676

The boiling-points of liquids increase as the pressure increases.

MELTING-POINTS OF VARIOUS SUBSTANCES.

The following figures are given by Clark (on the authority of Pouillet, Claudel, and Wilson), except those marked *, which are given by Prof. Roberts-Austen in his description of the Le Chatelier pyrometer. These latter are probably the most reliable figures.

Sulphurous acid	— 148° F.	Alloy, 1 tin, 1 lead.	370 to 466° F.
Carbonic acid	— 108	Tin	442 to 446
Mercury	— 39	Cadmium	442
Bromine	+ 9.5	Bismuth	504 to 507
Turpentine	14	Lead	608 to 618*
Hyponitric acid	16	Zinc	680 to 779*
Ice	32	Antimony	810 to 1150
Nitro-glycerine	45	Aluminum	1157*
Tallow	92	Magnesium	1200
Phosphorus	112	Calcium	Full red heat.
Acetic acid	113	Bronze	1692
Stearine	109 to 120	Silver	1733* to 1873
Spermaceti	120	Potassium sulphate	1859*
Margaric acid	131 to 140	Gold	1913* to 2282
Potassium	136 to 144	Copper	1929* to 1996
Wax	142 to 154	Cast iron, white	1922 to 2075*
Stearic acid	158	" gray	2012 to 2786 2228*
Sodium	194 to 208	Steel	2372 to 2532
Alloy, 3 lead, 2 tin, 5 bismuth	199	" hard	2570*; mild, 2687*
Iodine	225	Wrought iron	2732 to 2912
Sulphur	239	Palladium	2732*
Alloy, 1½ tin, 1 lead	334	Platinum	3227*

CHAPTER VII.

RATE OF COMBUSTION. DRAFT.

75. Introductory.—It has been previously observed that the calorific power of a fuel is a reasonably fixed quantity and is independent of the time taken to burn it (§§ 22, 23, 58, and 59). On the other hand the temperature of the fire depends on the number of pounds of fuel burned therein per hour (§§ 61 and 62), and the permitted dilution with excess of air (§ 30). It becomes of interest then to examine how many units of weight of a fuel, with a given heating effect, are usually burned per unit of grate-surface in a given time.

76. The Rate of Combustion.—The rate of combustion with English units is usually expressed by giving the number of pounds of fuel burned per square foot of grate-surface per hour. The table on page 96 gives a summary of practice and opinion upon this subject as respects steam-boilers.

An interesting comparison of tests recently made shows a tendency to regard 13 pounds per hour per square foot as representing prevalent American practice for stationary boilers on land. The use of higher pressures will be likely to increase this rate. The number of pounds of fuel which can be burned will be determined absolutely by the weight of oxygen or volume of air which can be supplied to it in a unit of time. It becomes a question of moment, therefore, to decide on the method of supplying the necessary air for combustion (§§ 24 to 27), and to provide a method to remove the

TABLE OF RATES OF COMBUSTION.

| Type of Boiler. | Pounds of Coal per Hour per Square Foot of Grate. | | Draft. | Authority. |
	Anthracite.	Bituminous.		
Land types..........	6–12	12–27	Chimney	Whitham
" "	....	65–80	Forced	Seaton
Marine types........	7–16	12–27	Chimney	Shock
" "	....	18–20	"	"
" "	....	20–30	Forced	Seaton
Cornish.............	....	4–6	Chimney	Rankine
Flue................	....	10	"	"
Factory	....	12–16	"	"
Marine	....	16–14	"	"
Locomotive........ ⟩	....	40–100	Forced	"
Torpedo-boat...... ⟨	....	60–125	"	....
Average...........	....	12–18	Chimney	....
"	....	60	Forced	....

products of combustion (§§ 28 and 29) which are not supporters of such combustion, but tend to extinguish it if not removed.

77. Draft for Combustion. General.—There are two generic ways of bringing air to a motive-power fire: it may be done by a mechanical apparatus, such as a fan or a steam-jet; or it may be done by means of the greater weight of cold air per cubic foot as compared with warm air, in a vertical column of such warm air enclosed within the walls of what is called a chimney. The first plan is known as mechanical or forced draft; the second is called natural draft or chimney-draft. The term natural draft is unfortunate, because the chimney is a simple machine, and both methods are mechanical, or equally dependent on a natural law. The term forced draft, on the other hand, should be restricted to cases in which the velocity of the air-current is made greater than it is possible to make it with a chimney of any ordinary or practical height. A mechanical appliance must be used in this case because the velocity is caused by a higher pressure

of the air where it enters the fire than can be secured by natural processes. The two alternatives are therefore natural or forced draft, having respect to the differences in pressure and velocity usual on the one hand; and the other division would be into chimney-draft or mechanical draft according as the current is produced by chimneys or by fans.

78. Chimney-draft. General.—It has long been a matter of common observation that heated air is lighter than the same bulk of cold air. In any volume of air either in enclosed spaces or in the open, the greater weight of the colder air will tend to draw it into the parts nearer to the earth, and in so doing it will displace the lighter warmer air and send it upward. If either the warmer or the cooler air or both are confined in tubes or ducts or flues, the movement of displacement becomes a definite flow through the flue or tube, and the action of gravity on the denser air becomes a means of moving the necessary weight or volume of air through the fuel to be burned. The chimney is the tube containing the lighter gas, at the bottom of which is the fire or source of heat. The column of cooler and heavier air outside of the chimney is the motor energy which sets the warm column in motion against the resistances of friction and the obstruction at the fire itself. The volume which the chimney will pass through its area will vary with its cross-section and with the velocity of flow through this cross-section. The velocity of flow will be fixed by the difference in weight of the hot column and of the cold column of equal height. This difference is fixed by the temperature outside the chimney and within it, which determines the density or weight per cubic foot of each column.

While it would appear, however, that the velocity increases with temperature, it must not be overlooked that the weight is decreasing with the higher temperature, and that weight of oxygen is really the important thing to be presented to the fire. There is therefore a certain temperature

of chimney-gas at which the greatest weight of oxygen will be passed through the fire. Below this the velocity is not high enough, and above this the weight per cubic foot is decreasing faster than the velocity is increasing. It is moreover of advantage not to have too high a chimney temperature made necessary to cause proper draft, because each unit of heat passing off in the chimney unnecessarily (§ 59) is a waste which should have been utilized, and that heat should have been transferred to the motor fluid in the form of heat energy instead of being wasted to create draft.

79. Theory of Chimney-draft by Péclet.—The most widely accepted theory of the action of the chimney was first elaborated by Péclet, and developed and quoted by Rankine
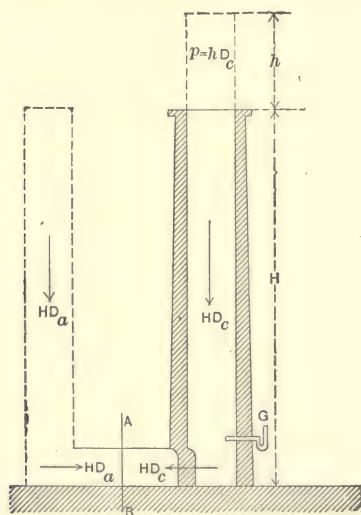
FIG. 13.

and other writers. His discussion can be made most easily to be understood by the conception of the chimney as an inverted siphon, with the fire-grate at the bend at the bottom. In Fig. 13 the hatched section represents the chimney,

and the dotted lines the siphon leg of cold external air. A
diaphragm A–B in the bend of the siphon will have unequal
pressures on its two sides if the legs are of equal length and
equal cross-section, because if D_a denotes the density of the
external air and its weight per cubic foot, and D_c denote the
density of the warmer lighter chimney-air, then HD_a acts on
one side, and the less HD_c on the other.   To equalize the
pressures an extra effort must be exerted on the lighter leg
to balance the heavier, so that an extra length of column of
hot air of unknown height h, and having a density D_c, must
exert a pressure $p = hD_c$ to effect this balance.   Or, since

$$HD_a = HD_c + p,$$

we can write

$$p = HD_a - HD_c$$

as the pressure exerted on the diaphragm or on a film of air
at the base of the siphon and which causes the flow when
there is no balancing pressure at the top of the chimney.
But since $p = hD_c$, the height of the column of hot gas
will be

$$h = \frac{H(D_a - D_c)}{D_c},$$

and the question of the values of these two densities is a
question for observation or calculation.   At 32° F., D_0 for
air is .0807, and by reducing D to 32° F. the value for h
can be found in feet, or more conveniently the expression can
be transformed to read in absolute temperatures instead of
densities by the relation that the densities will be inversely
as the temperatures, so that

$$\frac{D_c}{D_0} = \frac{T_0}{T_c}.$$

But the chimney-gas is a mixture, and not a constant or
permanent gas.   An accepted value for its ordinary density

at $32°$ F. is .08424, which is derived from an averaging of many analyses and experiments which give for such gases:

Carbonic acid 10 per cent, weighing at $32°$..... .12344
Nitrogen   79 "   .   "   " " ..... .07860
Oxygen   11 "   "   " " ..... .08926

Multiplying the per cent of each by its weight, we have:

Weight of CO_2....................... .01234
 "   " N...................... .06209
 "   " O....................... .00981

Total....................... .08424

If the composition of the gases differs from the above assumption as determined by analysis or otherwise, additional data are given in the following table:

	Specific Volumes.	Specific Heat in Gaseous Condition.	Density or Weight of One Cubic Foot.
Hydrogen	178.881	3.4090	0.00559
Oxygen	11.2070	0.2175	0.08928
Nitrogen	12.7561	0.2438	0.07837
Carbon dioxide..................	8.10324	0.2169	0.12341
Carbon monoxide.................	12.81	0.2450	0.07806
Water	....	0.4805	....
Air	12.3900	0.2375	0.08071
Ash.............................	....	0.2	....

Substituting, then, for D in the formula for height the expression

$$D_c = .08424\frac{T_o}{T_c},$$

the expression for that height becomes

$$h = H\frac{\left[.0807\frac{T_o}{T_a} - .08424\frac{T_o}{T_c}\right]}{.08424\frac{T_o}{T_c}},$$

which becomes by performing the operations

$$h = H\left(.96\frac{T_c}{T_a} - 1\right).$$

The velocity in feet per second caused by a height h in feet will be denoted by $v = \sqrt{2gh}$; the volume V per second if the cross-section be denoted by A square feet will be Av, and becomes

$$V = Av = A\sqrt{2gH\left(\frac{.96\,T_c - T_a}{T_a}\right)}$$

for the temperature of 32° F. If it be required to burn W pounds of coal per second, and KW cubic feet of gas at 32° F. be the result, we shall have an equation for W, since $V = KW$.

$$W = \frac{A\sqrt{2gH\left(\frac{.96\,T_c - T_a}{T_a}\right)}}{K}$$

as the theoretical pounds of coal which will be burned by a chimney of height H and area A when the resistances to flow of air and gas are disregarded.

80. Discussion of Péclet's Theory of Chimney-draft.— Péclet developed a later theory in which the dynamic energy for the flow of air to the furnace was a head in feet expressed in terms involving the cold gas or external air. He also developed an expression for the velocity of flow, starting from the general expression for the relation between the head in a pipe and the flow which it produces in the case of a liquid. A form for this is

$$h = \frac{v^2}{2g}\left(1 + K + K_1 + \frac{fl}{m}\right),$$

which in Peclet's form appears as

$$h = \frac{v^2}{2g}\left(1 + G + \frac{fl}{m}\right).$$

In this equation h is the head; g is the acceleration due to gravity; f is the friction against sides of pipe or duct or flue; K and K_1 or G, which combines them, are coefficients to express the resistances offered by bends, elbows, valves, and fittings in hydraulic work and by the grates, tubes, and damper in boiler-furnace work; l is the length of the pipe or gas-passage; and m is the ratio of area of cross-section to the perimeter, called the hydraulic mean depth. For square or round flues m will be one fourth of the side or diameter, since $\dfrac{b^2}{4b} = \dfrac{b}{4}$ for a square flue, and $\dfrac{\pi r^2}{2\pi r} = \dfrac{r}{2} = \dfrac{D}{4}$ for a round one. Péclet's value for G he puts at 12 for cases where 20 to 24 pounds of coal are burned per hour, and for f his value is 0.012 for surfaces covered with soot. Hence his formula becomes

$$h = \frac{v^2}{2g}\left(13 + \frac{0.012l}{m}\right),$$

whence the expression for volume per second with a height H would appear

$$V' = Av' = A\sqrt{\frac{2gH\left(\dfrac{.96\,T_c - T_a}{T_a}\right)}{13 + \dfrac{0.012l}{m}}}.$$

The uncertainty as to coefficients; the fact that it is not true that $v = \sqrt{2gh}$ for a flow of a gas which undergoes any notable change in pressure or temperature, and the chimney problem introduces both; the fact that the chimney temperature T_c is not constant from top to bottom; and the necessity for the assumptions of area and temperature and velocity

before a height can be worked out, have thrown designers upon the data of experience rather than upon the foregoing calculations.

The Péclet formula, however, possesses this interest. Since the velocity of the gas in the chimney increases as the square root of the height of the dynamic column, and therefore with $\sqrt{.96\,T_c - T_a}$ when the external-air temperature is fixed, and since the density is inversely proportional to the temperature in the chimney, the weight discharged will be proportional to

$$\frac{\sqrt{.96\,T_c - T_a}}{T_c},$$

which becomes a maximum when

$$T_c = \frac{2\,T_a}{.96} = \frac{25}{12}\,T_a = 2\tfrac{1}{12}T_a;$$

or the greatest weight will be discharged when the absolute temperature within the stack is $\frac{25}{12}$ of the absolute temperature of the external air. That is, if the external air be at 62° F., or an absolute temperature 522°, the temperature within the chimney for a greatest weight of gas flowing should be 522 × $\frac{25}{12}$ or 1087° absolute, or 626° F. This explains the usual preference for temperatures around 600° F. in ordinary boiler-stacks. This is about the temperature of melting lead. On the other hand, for many metallurgical purposes a higher temperature in the stack is a necessity, and a greater velocity than is usual in steam-boiler practice. When this maximum temperature prevails $h = H$; or the extra column of hot gas has a height equal to that of the original chimney, and the density of that gas is one half of that of the external air. The formula also indicates that chimneys draw best with cold air outside and at high barometric pressures.

81. Some Accepted Chimney Formulæ and Data.—Mr. Wm. Kent in 1884 proposed a formula based on successful practice and on the idea that the effective area of a chimney was less than its gross area by a dead-space of two inches radially from each wall of a square chimney or all around a round one. This idea, if A be the gross area expressed in square feet, and E the effective area, will make:

For square chimneys

$$E = D^2 - \tfrac{8}{12}D = A - \tfrac{2}{3}\sqrt{A}.$$

For round chimneys

$$E = \pi(D^2 - \tfrac{8}{12}D) = A - 0.592\sqrt{A}.$$

This is so nearly the same for both that it can be written

$$E = A - 0.6\sqrt{A}.$$

Since the power of a chimney varies both as the square root of its height at best temperature conditions and as its effective area, it can be written that

$$\text{H.P.} = E\sqrt{h} \times C,$$

in which C is a constant to be determined from successful practice. A boiler horse-power is assumed to be equivalent to an evaporation of 30 pounds of water per hour (§ 101). Assuming 5 pounds of coal per horse-power per hour to take account of poor conditions, and observing the number of pounds of coal which a successful chimney will take care of, an acceptable value for C is found to be $3\tfrac{1}{8}$. Hence

$$\text{H.P.} = 3.33E\sqrt{h} = 3.33(A - .6\sqrt{A})\sqrt{h},$$

which can be written also

$$E = \frac{0.3\,\text{H.P.}}{\sqrt{h}},$$

when the quantities of the second member are the known data.

A series of observations by Morin & Tresca from French

practice have resulted in the following table, which is a very safe guide. The grate is eight times the chimney cross-section.

Heights in ft. of chimney........	20	25	30	35	40	45	50	55	60	65
Lbs. per hour per sq. ft. grate....	7.5	8.5	9.5	10.5	11.6	12.4	13.1	13.8	14.5	15.1
" " " " " " chimney.	60	68	76	84	93	99	105	111	116	121

Heights in ft. of chimney........	70	75	80	85	90	95	100	105	110
Lbs. per hour per sq. ft. grate....	15.8	16.4	16.9	17.4	18.0	18.5	19.0	19.5	20.0
" " " " " " chimney.	126	131	135	139	144	148	152	156	160

A simple formula by Thurston agreeing quite closely with the above table is

$$\text{Rate of combustion} = 2\sqrt{h} - 1,$$

in which h is the height in feet.

Other designers have aimed to deduce formulæ from practice which should take account of the prevalent resistances in grates and fires with different grades of fuel, introducing the results of tests into formulæ as coefficients. But successful practice of others will remain the preferred guide. Sectional-boiler practice using water-tubes has deduced the following diagram (Fig. 14) to represent the draft in inches of water

DIAGRAM OF DRAFT AND CAPACITY OF CHIMNEY.
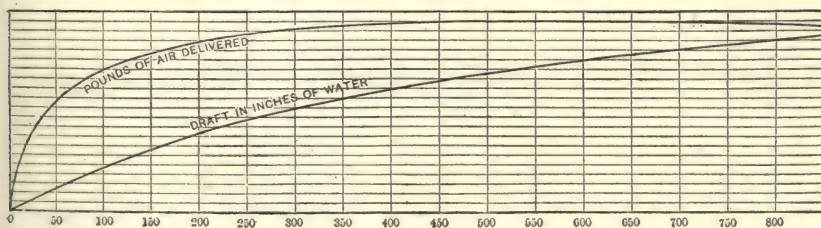
Fig. 14.

(§ 83) corresponding to any number of pounds of air delivered when the chimney is 100 feet high and the external air is at 60°, as well as the maximum rendition-point between 500° and 600°.

Chimneys over 150 feet in height are rarely justified; but

250 feet of height may be compulsory in towns to carry off gaseous or noxious products without possibility of nuisance. The following table represents conservative data:

Pounds of coal consumed per hour.....	Up to 100	500	1000	2000	3000	4000	5000
Height in feet...........................		60	100	120	140	160	180	200

Several smaller chimneys are often used instead of one large one, where location does not compel great height, with considerable economy.

Fine anthracite coal needs a higher stack than good bituminous coal, both on account of the grate resistance and the lower temperature of the gases, and wood requires less than either of the other two.

Tallest chimneys of record are:

Townsend's Chemical Works, Glasgow......... 468 feet
Hallsbruckner Hütte, Saxony................ 460   "
Metropolitan Street Railway Co., New York... 353   "
Omaha & Grant Smelting Co., Denver........ 352   "
Clark Thread Co., Newark, N. J............. 335   "
Amoskeag Mills, Manchester, N. H.......... 250   "
Narragansett E. L. Co., Providence......... 238   "
Maryland Steel Co., Sparrows Pt., Md....... 225   "
Passaic Print Works, Passaic, N. J.......... 200   "
Edison Electric Light Co., Brooklyn (two).... 150   "

82. Cross-section of Chimney.—The weight of chimney-gas moving per second through the fire is conditioned both upon velocity and cross-section, which for a fixed quantity of gas can vary inversely as each other. Too large a cross-section causes the chimney to draw badly because the lower velocity of the gases permits eddies from back-draft; the gases are cooled by contact with the material of the chimney, and the chimney is unnecessarily costly to build. Hence a sort of empiric standard makes the cross-section of the chimney to be one eighth of the area of the grates burning coal for it. This can be shown to be ample for any usual assumptions or normal velocity; for, if an area of one square foot be

taken, and a temperature for maximum output $v_1 = \sqrt{2gH}$, and if H be taken at 64 feet of height for illustration,

$$v_1 = 64 \text{ cubic feet per second}$$
$$= 64 \times 3600 = 296,000 \text{ cubic feet per hour.}$$

Suppose 20 pounds of coal burned per hour per square foot of grate, and 300 cubic feet of air per pound of coal; then $20 \times 300 = 6000$ cubic feet of air at 62° will be required per square foot of grate. At 626° F. in the chimney this air will have twice its volume at 62°, since

$$V_c = V_a \frac{T_c}{T_a} = 300\tfrac{1086}{521};$$

whence

$$V_c = 12,000 \text{ cubic feet,}$$

which, if multiplied by the assumed relation of chimney $1 :$ grate $8 = 96,000$, is only about $\frac{96000}{296000}$, or one third of what the chimney of only 64 feet high will take care of per foot of area of cross-section.

The friction becomes greater if the chimney be too small, and plants are usually enlarged after some years of use. Hence, although this one-eighth value is large, it is usually best not to pass much below it in small plants. Possible excess of area is corrected by partly closing the damper in the flue to the stack.

An ingenious designer has proposed to use the dead area of the Kent formula, or the back-draft area in the above discussion, as a passage to bring preheated air down the stack so as to introduce it below the fire and avoid the consumption of fuel required to raise this air to fire-temperature.

83. Draft-gauges.—It is usual to observe the pressure prevailing at the base of the stack after the resistances caused by the fire, grate, damper, and flues have been encountered, and to call this pressure that which causes the flow in the

siphon. In the foregoing discussion it was given in feet of
head of hot gas (h), but can be transformed into pounds per
square foot by multiplying (h) by the weight of a cubic foot
of the gas (D_c), or

$$p = hD_c,$$

as shown in § 79. The velocity is then worked out from the
pressure by finding the value for h in feet when p and D_c are
observed, and by calling $v = \sqrt{2gh}$.

The most usual form of draft-gauge is a U tube or in-
verted siphon of water, of which one leg is connected to the
chimney-base by a convenient pipe-connection, and the other
is open to the atmosphere (Fig. 15). This gauge gives a
reading in inches of water, and the relation between the two
units is found as follows: since 1 cubic foot of water exerts a
pressure per square foot of 62.5 pounds, hence 1 inch exerts
$\frac{62.5}{12} = 5.2$ pounds per square foot, and $\frac{1}{5.2} = 0.192$. If
then D_a and D_c are densities as before, and we have values

$$D^a = .0807\frac{T_0}{T_a} \quad \text{and} \quad D_c = .084\frac{T_0}{T_c}.$$

Then, if the force of the draft be denoted by f,

$$f = .192H(D_a - D_c);$$

or, when the values are substituted,

$$f = .192H\left(\frac{39.79}{T_a} - \frac{41.41}{T_c}\right) = H\left(\frac{7.64}{T_a} - \frac{7.95}{T_c}\right).$$

A similar calculation under the condition of maximum
quantity of draft, with 60° F. outside and 600° in the chim-
ney, gives

$$f \text{ (in inches of water)} = .0073H.$$

The U tube or siphon-gauge is not sensitive to very slight
differences of pressure, and many improvements have been

suggested, such as using fluids of different specific gravities in the two tubes (Fig. 16); using a large surface to receive the light pressure, while reading the change of pressure by a hook-
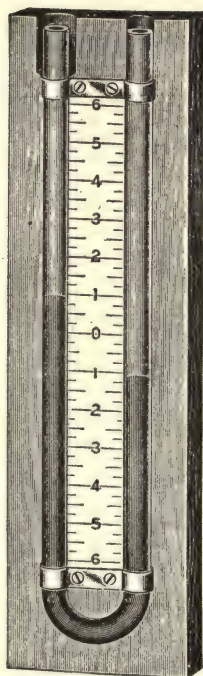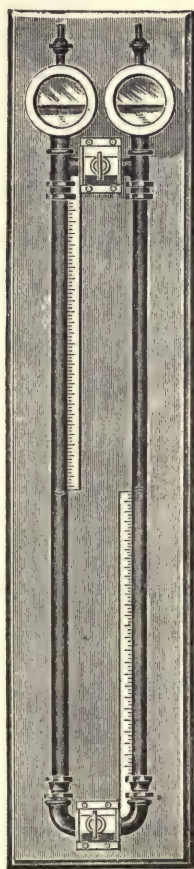
Fig. 15.   Fig. 16.

gauge (Fig. 17); the use of the gas-holder principle, and many other arrangements. Five to seven inches of water is as high a reading as is secured with the intense forced draft of locomotives or torpedo-boat practice; two inches is more

usual when the air is forced into the ash-pits from **below the** grates; a fraction of an inch (from $\frac{5}{10}$ to $\frac{6}{10}$) is all which can
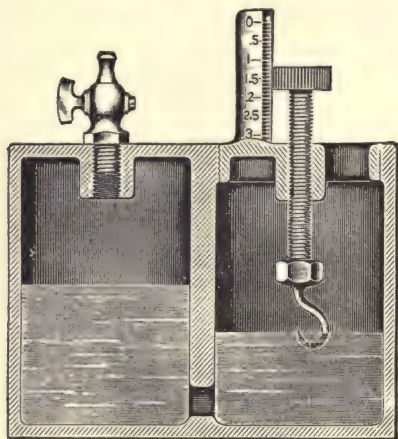
FIG. 17.
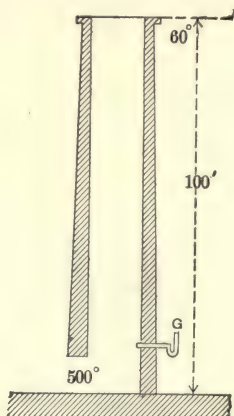
FIG. 18.

be usually counted on with ordinary chimneys, as by the following illustrative example:

What would the water-gauge G (Fig. 18) read when D_1 for outer air $= .0764$, and D_2 in the chimney $= .0413$?

$$\frac{D_1}{D_0} = \frac{T_0}{T_1}.$$

$$\therefore \quad D_1 = \frac{D_0 T_0}{T_1}.$$

$$\frac{D_2}{D_0} = \frac{T_0}{T_2}.$$

$$\therefore \quad p = h(D_1 - D_2)$$
$$= 3.51 \text{ lbs. per square foot,}$$

and $\quad p_1 = 3.51 \times 100 \times .192$
$\qquad = .67$ inch of water,

or $\qquad 3.51 \times 1 = p_1 \times 62.5.$
$\qquad\qquad p_1 = .056$ foot
$\qquad\qquad\quad = .67$ inch.

84. Flue-gas Analysis.—It has long been appreciated that much valuable information can be derived from an analysis of the flue-gases in the chimney, at any rate so far as the presence of uncombined oxygen, carbon monoxide, and carbonic acid are concerned. An excess of oxygen means a loss by dilution and a lowered fire-temperature; any quantity of carbonic oxide means a waste of carbon unconsumed because there is available heat in it which has not been liberated by combustion.

The remainder of the flue-gases is commonly assumed to be nitrogen, but it includes unburned hydrocarbon, if there be any, and steam or vapor of water. The Orsat apparatus is the most used in analyzing flue-gases and is illustrated in Fig. 19. P''', P'', and P' are pipettes containing, re-
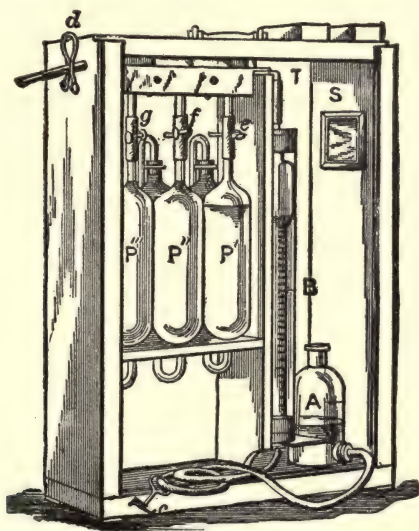
FIG. 19.

spectively, solution of caustic potash to absorb carbon dioxide, pyrogallic acid and caustic potash to absorb oxygen, and cuprous chloride in hydrochloric acid to absorb carbon monoxide.

At d is a cock to control the admission of gas to the

apparatus; at B is a graduated burette for measuring the volumes of gas; and at A is a pressure-bottle connected with B by a rubber tube to control the gases to be analyzed. The pressure-bottle is commonly filled with water, but glycerine or some other fluid may be used when, in addition to the gases named, a determination of the moisture or steam in the flue-gases is made.

The several pipettes P', P'', and P''' are filled to the marks g, f, and e with the proper reagents, by aid of the pressure-bottle A.   With a three-way cock to open to the atmosphere, the pressure-bottle A is raised till the burette B is filled with water to the mark m; communication is then made with the flue, and by lowering the pressure-bottle the burette is filled with the gas to be analyzed, and two minutes are allowed for the burette to drain.   The pressure-bottle is now raised till the water in the burette reaches the zero mark and the clamp c is closed.   The valve in the pipe to the flue is now opened momentarily to the atmosphere to relieve the pressure in the burette.   Now open the clamp c and bring the level of the water in the pressure-bottle to the level of the water in the burette, and take a reading of the volume of the gas to be analyzed; all readings of volume are to be taken in a similar way.   Open the cock g and force the gas into the pipette P''' by raising the pressure-bottle, so that the water in the burette comes to the mark m.   Allow three minutes for absorption of carbon dioxide by the caustic potash in P''', and finally bring the reagent to the mark a again.   In this last operation, brought about by lowering the pressure-bottle, care should be taken not to suck the caustic reagent into the stop-cock.   The gas is again measured in the burette, and the diminution of volume is recorded as the volume of carbon dioxide in the given volume of gas.   In like manner the gas is passed into the pipette P'', where the oxygen is absorbed by the pyrogallic acid and caustic potash; but as the absorption is less rapid than was the case with the carbon monoxide,

more time must be allowed, and it is advisable to pass the gas back and forth, in and out of the pipette, several times. The loss of volume is recorded as the volume of oxygen. Finally, the gas is passed into the pipette P', where the carbon monoxide is absorbed by cuprous chloride in hydrochloric acid.

The solutions used in the Orsat apparatus are:

P'''. Caustic potash, 1 part; water, 2 parts.

P''. Pyrogallic acid, 1 gram to 25 cc. of caustic potash.

P'. Saturated solution of cuprous chloride in hydrochloric acid having a specific gravity of 1.10.

These reagents will absorb per cubic centimeter:

P'''. Caustic potash absorbs   40 cc. of CO_2;
P''. Pyrogallate of potash absorbs 22 " " Oxygen;
P'. Cuprous chloride absorbs   6 " " CO.

Improvements in the Orsat apparatus and its manipulation have been made by Hempel, Carpenter, Hale, and others, and the student is referred to Hempel's treatise for further detail.

85. Stability and Structure of Chimneys.—The chimney and a proper foundation for it belong rather to structural engineering than to a treatise upon heat, and it would divert from present purposes to discuss these questions at length. Wind-pressure is not likely to reach 55 pounds per square foot of flat surface; and the chimney may be viewed as a cantilever loaded uniformly with this load. In brick structures this must never produce tension on the windward side, when compounded with the resistant weight of the bricks, which will range from 100 to 130 pounds per cubic foot; nor on the compression side must the stress exceed 8 tons to the square foot, which the brick should be able easily to withstand. That is, if h be the height in feet, d the average breadth, and b the breadth at the base, there must be equilibrium between W, the weight of the chimney in pounds,

and the quantity $C\dfrac{dh^2}{b}$.   In the latter, the coefficient C is a factor for wind-pressure per square foot of area.   It is 56 for a square chimney, 35 for an octagonal, and 28 for a round chimney.   A brick chimney so proportioned will withstand any gale likely to be experienced.   It will appear, however, that a chimney from these causes and the concentration of weight on a small area is a structure particularly liable to unequal settling of its foundations.   The latter, therefore, should receive most careful attention from a competent designer of foundations, and should be laid by experienced persons.   Natural and undisturbed soil will carry one ton per square foot; loam, compact sand, or hard-pan can carry two tons per square foot.   Where natural foundations cannot be had, piling and other artificial methods are to be resorted to.

With respect to their structure, chimneys may be grouped into—

(1) Brick.

(2) Steel or iron shell, brick-lined.

(3) Skeleton iron and brick.

Brick chimneys are round, square, octagon, or star-shaped.   Circular section seems best, as lighter, stronger, and more shapely.   English rule is, base equals one tenth of height; the batter or taper in American practice is from one sixteenth to one quarter inch to the foot on each side.   One in thirty-six is the English standard.

The upper 25 feet is one brick thick (8″ or 9″); thickness increases by one half brick per 25 feet.   If the diameter exceeds 60 inches, begin at top with one and a half bricks.

An inner lining or core, detached from the wall proper and running either nearly to top or over 50 or 60 feet up, prevents expansion from cracking the walls.   It need not be fire-brick all the way up, or even further than one half.   The core is made of tangent-laid brick, with an occasional header to guide the core by the wall.

Another practice is to make a 100-foot chimney in three sections: first, 20 feet high, 16 inches thick; second, 30 feet high, 12 inches thick; third, 50 feet high, 8 inches thick. Core in three sections of 12, 8, and 4 inches thick, respectively.

The top of a chimney is exposed to weather and frost and snow, melting and freezing. There should be a cast-iron cap, or a stone, to protect the top edge of the brick. Large moulded terra-cotta or fire-clay blocks are also used, clamped and dowelled together.

Cylindrical steel chimneys of riveted plate steel, secured by a flare in the lower 10 to 25 feet to a cast-iron base-plate, which again is anchored by heavy foundation-bolts to a masonry foundation, require no guy or stay ropes and are 35 to 50 per cent cheaper than a brick stack. They take less room, are strong and safe, and no air leaks in to cool the gas. They are brick-lined part way or all the way up. They must be kept painted.

Stacks when not anchored to foundations by bolts, and all light and unlined stacks, require to be stayed by guys of wire rope. They are attached opposite the centre of effort of the winds, at two thirds of the height; are usually four in number, the first being led in the direction of the most violent wind, and each guy of a cross-section in square inches one thousandth of the exposed area in feet.

Skeleton chimneys have been put up by iron-works, but have no advantage over steel cylinders, and for many reasons are not as good. Brick is built in between uprights of rolled iron, which are banded by flat rings on the outside.

Access should be permitted to the chimney at its base through a proper door either in the flue or in the foundation of the chimney, and it is best that a ladder on the outside of the chimney should give access to its top. In a square chimney this ladder can be made by bars let into two walls at a corner. Figs. 20, 21, and 22 show chimney constructions and the proportions which have been found satisfactory,

FIG. 21.
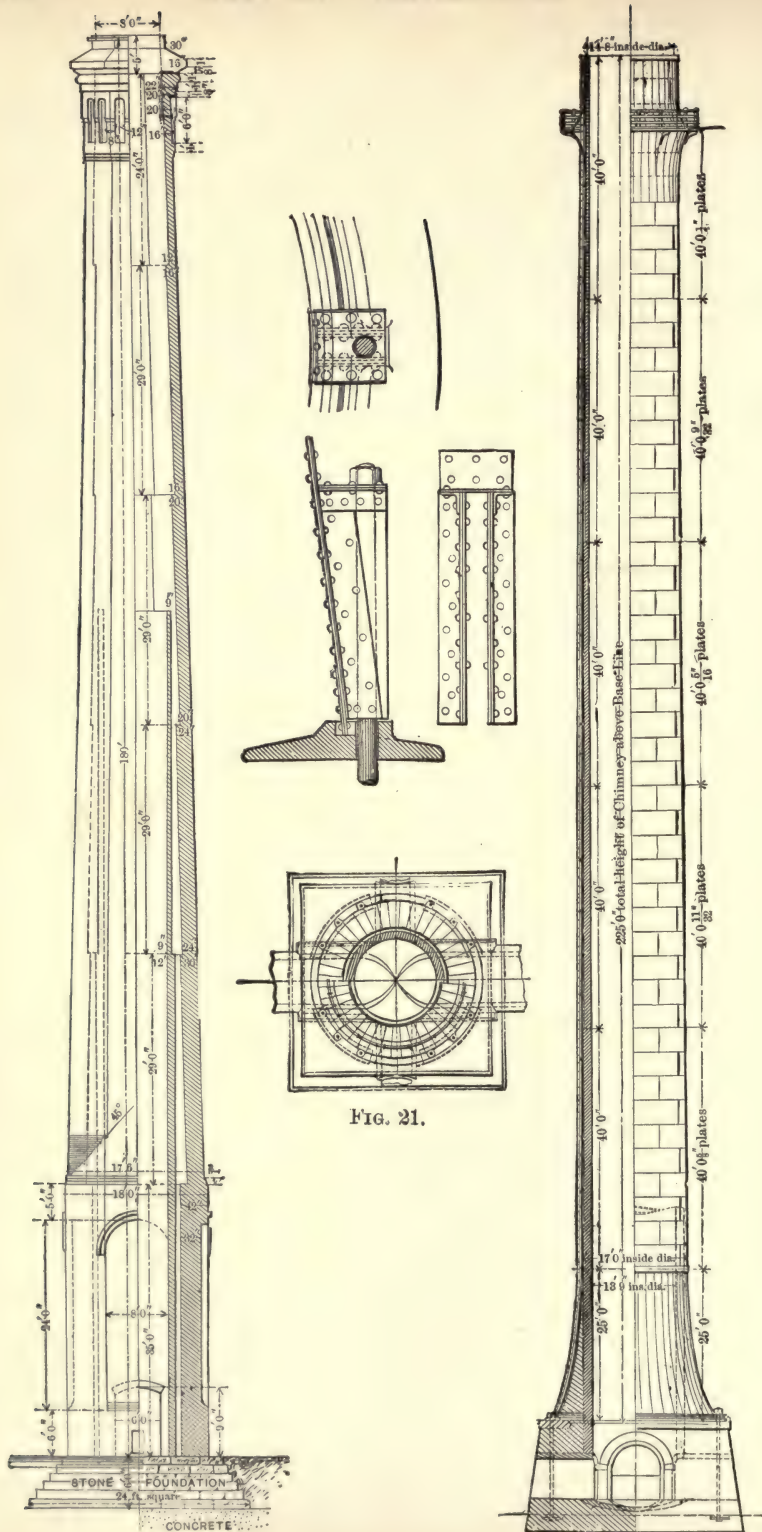
according to which the thickness may be reduced as the chimney attains height.

86. Artificial or Mechanical or Forced Draft.—It has been already pointed out (§ 72) that a movement of the air for combustion might be mechanically produced by a proper appliance for this purpose.

A calculation of efficiencies shows that for heights of chimneys such as are ordinarily used the mechanical methods of securing draft are the more efficient, so that it becomes a question of consideration whether the necessary air for combustion shall be furnished by a costly chimney or group of them, or by a continuously running machine of some different type.   Artificial draft can be secured by two general methods.   The first type is that made familiar in locomotive practice, in which a rapid motion is given to the air to draw it out of the smoke-box so that the reduction of pressure within the latter shall cause a flow through the grates, fire, and tubes to equalize this rarefaction.   This is called the induced-draft system, and as applied when fans are used, as in steamship practice, is illustrated in Fig. 25.

The other plan is to cause a pressure of air in the ash-pit below the grate-bars so that the air will flow up through the fire, the setting, and flues by the excess of pressure which prevails in the ash-pit.   This is called the forced-draft system, and is becoming more usual in high-speed marine practice.   The movement of the air can be produced either by means of a steam-jet inducing a current of air to flow, or fans or blowers either of the centrifugal or positive type may be used.   If the first or aspirating principle is used, the products of combustion must pass over the aspirating appliance. These gases are hot and possibly corrosive.   The heat makes lubrication difficult, and almost excludes the use of apparatus where lubrication must be provided unless all bearing-surfaces can be without the flues which carry the gas.   Protection against corrosion can be secured if proper trouble is taken,

but where this is not guarded against the apparatus deteriorates rapidly.   The forcing system has the fresh cool air pass through the forcing appliance, and has furthermore the advantage of maintaining a higher tension within the setting
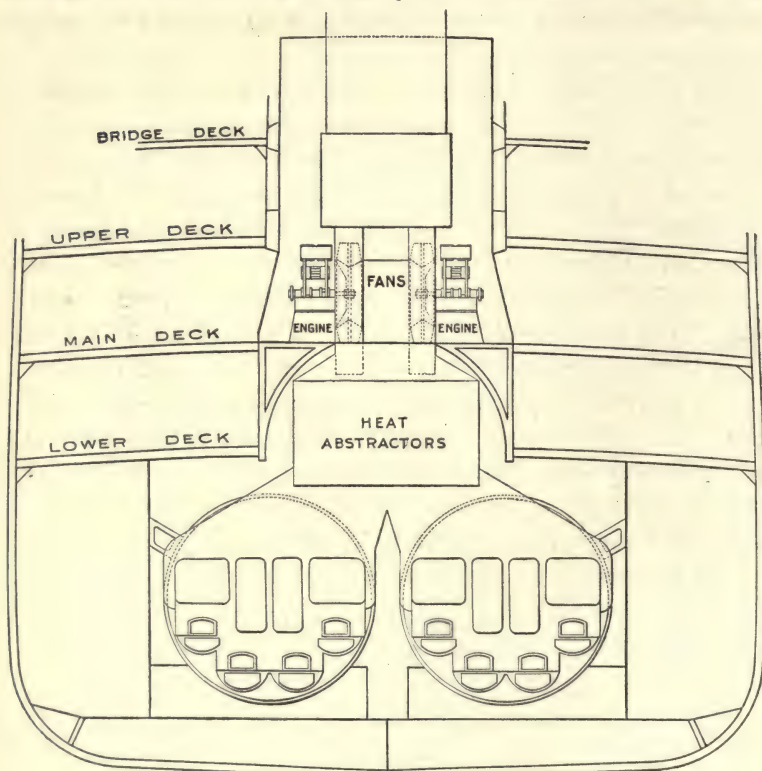
Fig. 25.

than prevails outside of it, so that there is little or no tendency for cool air to leak through cracks or porous brick-work into the gas-currents.   This is a difficulty present where the draft is done by aspiration.   On the other hand, the pressure system makes a hot and gassy fire-room if there are places where gas can escape through cracks, doors, or elsewhere from within the setting into the room.   Fig. 26 shows Mr.

Jno. C. Kafer's closed ash-pit system, similar to that on the U. S. S. *Swatara* and *Kearsarge*. Since combustion is more efficient the denser the air used to effect it, the pressure
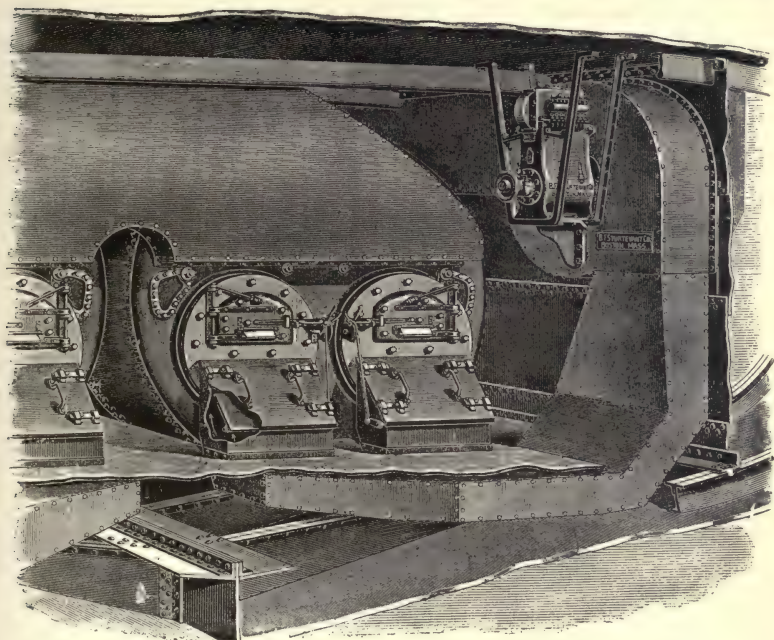
FIG. 26.

system offers an advantage from this point of view, as compared with natural draft or the aspiration system.

87. Advantages of Artificial Draft.—It is to be said in favor of natural or chimney draft that, when the chimney is once built and paid for, the draft-machine costs nothing to run except the heat which is used for this purpose, and it undergoes little or no deterioration with use. Furthermore, in cities the necessities imposed upon the power plant to carry the products of combustion high enough up to create no nuisance in its neighborhood compel a height and cost of chimney which makes the consideration of artificial draft

unnecessary, since the high chimney must be there in any case. Again, where the plant is so large that the cost of the draft-machine becomes considerable, or, what is the same thing, the cost of the expensive chimney becomes distributed over a large number of horse-power units, the advantages of artificial draft are not so apparent.

Artificial draft, on the other hand, offers the following advantages:

(1) The rapidity of combustion in the fire-box is not limited by atmospheric conditions. With a demand for high steam-pressure and great capacity in a limited space the forced draft is a necessity, as in war-ship practice.

(2) It is possible to increase the evaporative capacity of a given plant without other change than the velocity of the draft-machine. This increase may be either permanent or to meet sudden demands for steam, such as occur in street-railway practice at busy hours. With natural draft the chimney must be designed to meet the maximum requirement, and will be partly shut off at other times.

(3) It is possible to burn inferior, cheaper, and smaller sizes of fuel with artificial draft, because a high pressure can be maintained which will force the necessary air through a compact body of fuel.

(4) The draft arrangements are more portable than chimneys can be.

(5) The plant is more flexible for changes in quality or size of fuel, and the desirable thickness of fuel-bed on the grates. Grate-bars can be altered more easily if this should be desirable.

(6) Where high stacks are not made necessary the cost which they entail is avoided, or is obviated by a less cost of the draft-machine. The troublesome settling of massive stacks is avoided when foundations are difficult or defective.

(7) Leakage of air into the setting does not occur with forced draft on the pressure system.

88. Disadvantages of Artificial Draft.—The objections to be raised against the artificial draft are:

(1) The running cost of the machine. While it takes less coal than the chimney to do a given work, the fuel is not the only expense where an engine must be run, consuming oil and other supplies, calling for repairs and supervision, and the expense of the latter may be considerable.

(2) The artificial-draft machine occupies space which can often be ill spared.

(3) Running machinery, and particularly that at high speed such as most draft appliances demand, is rarely silent, is often noisy, and is liable to breakdowns which compel it to stop.

It will be seen that chimney-draft is not liable to these disadvantages.

The machine for causing the draft may be a centrifugal fan driven either by its own directly coupled engine or by a detached engine, or a revolving shaft, or by means of an electrical motor. The positive blowers will be driven by belts, or their own direct-coupled engine or motor, whether used for pressure or suction methods, and the steam-jet, which is the third appliance, requires no moving machinery when used in either system. It will be seen that each of these offers some advantages and disadvantages of its own. The fan method, if driven by belting, increases the running cost; and if electric current must be generated, the cost of its transformation must be considered. The steam-jet plan occupies very little space and is cheap to buy in the first instance. It is, however, wasteful of steam as compared with the other systems, and is in most cases too noisy. If used as a forcing system, the steam passes through the fire and is objectionable. If used as a suction system, the steam goes out with the products of combustion and does no harm.

The methods which have been used in marine practice to secure the necessary forced draft are either the closed ash-

pit system, the closed fire-room system, or the induced-draft
system.   The combination of closed ash-pit system with the
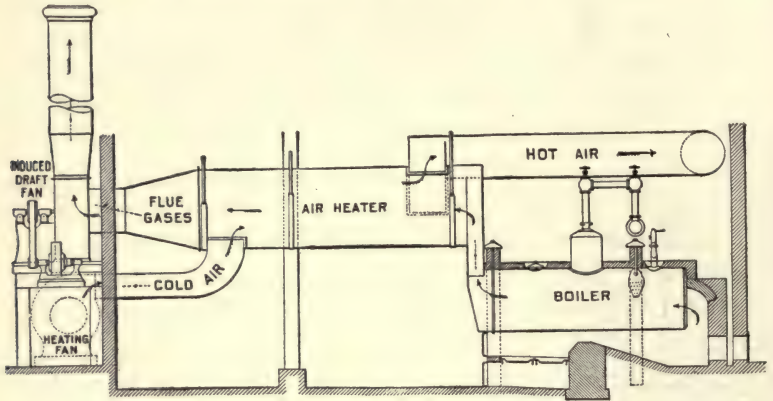
Fig. 27.
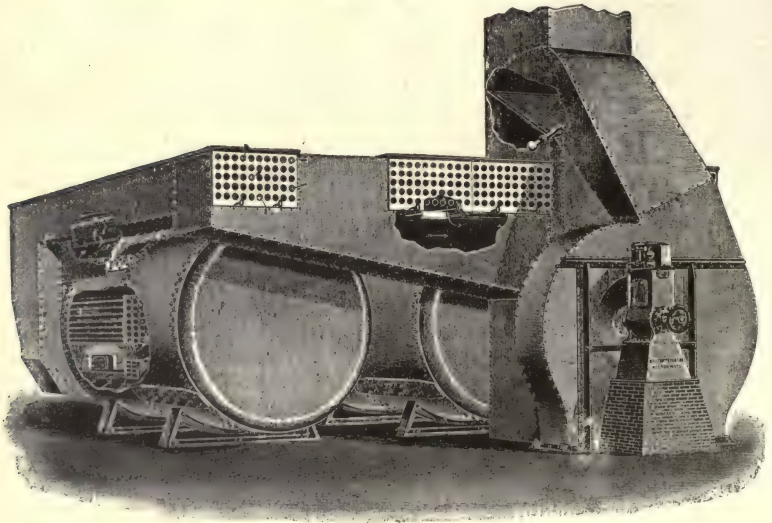
Fig. 28.

induced-draft system enables preheating of the air to be easily
done before it enters the ash-pit.   Figs. 27, 28, and 29 show
typical stationary arrangements.

89. Smoke-prevention.—The preceding discussion on the liberation of heat from a fuel for motive-power purposes would not be complete without a reference to the loss of energy which occurs when combustible carbon passes out with the products of combustion, and without having undergone complete oxidation at the desired point. When this carbon goes off as carbon monoxide, the loss is that made
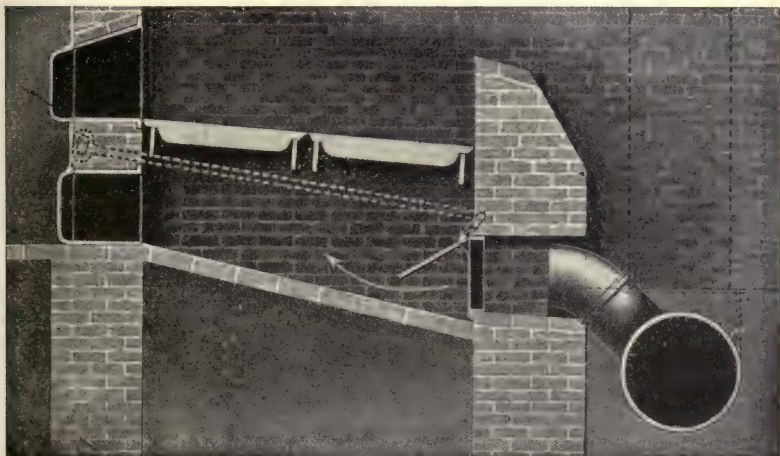
Fɪɢ. 29.

manifest in § 24. When incandescent solid carbon fails to meet oxygen under favorable conditions for its union with it, the extinction of the glowing particles forms them into lamp-black or soot, which particles color the products of combustion, and cause them to darken the air and to defile the surfaces which they touch. A smoke, in its exact sense, is a current of products of combustion from a fire, in which the otherwise colorless gases carry finely divided particles of black carbon. This carbon resulting from incandescence which has ceased is practically incombustible at ordinary heats. It could have been burned, however, if the union with oxygen had taken place while the carbon was in the nascent or favor-

able state of its first incandescence, and the effort of the
designer and manager of the combustion must be directed to
keep up the gases to the temperature of the ignition of the
carbon, and with a full supply of oxygen at sufficient temper-
ature to satisfy the carbon.   Pure hydrogen combustions are
normally smokeless, because of the absence of solid matter
in the flame.   Such flames are usually non-luminous for the
same reason.

The various methods for smoke-prevention have been
grouped under the following heads:

(1) The supply of excess of air by steam-jets, inducing
currents which they warm, and supplying excess of warm air
above the fire and behind the bridge-wall.   The difficulty
with these has been that, after distillation of the gas is com-
pleted, after a charge of fresh fuel is thrown on the fire, this
excess of air is not needed, and the products of combustion
are cooled by the diluting oxygen.   Attempts have been
made to correct this by graduating the supply of fresh air by
chronometric or other appliances, so that the excess should
be cut off after such an interval as is usually needed for the
first distillation of gas.

(2) By the coking methods of firing.   By these plans a
large dead-plate was used, so that the gases should be distilled
off from the fresh fuel before its combustion was really begun
on the grate-surface proper, and when the coking was com-
plete only fixed carbon remained to burn on the grate-surface
proper when pushed back.   The gas distilled from the fuel
on the dead-plate passed over the hot fire, and was so
warmed that it was ready to combine and burn.   Alternate
firing of the two sides of the furnace, or the use of two fur-
naces delivering into a common combustion-chamber which
were fired alternately, belong to this same class.

(3) The methods belonging to the principles of mechan-
ical stoking are smoke-preventing methods in that each part
of the fire always remains in the same condition, and the fresh

coal which distils off gas is received in the coolest part of the grate, and passes to the hotter sections only after the volatile matter has been distilled off and burned in passing over those hottest portions.

(4) Gas- and oil-firing are smoke-preventing methods, since when properly done the combustion ought to be complete, and no carbon should pass out of the setting except in the form of carbonic acid. It is to this group that those settings belong in which the actual combustion of the fuel containing volatile matter is done in a separate furnace and away from contact with the boiler. This makes a relatively smokeless and efficient apparatus, and will answer with coals which cannot be economically burned in any other way.

(5) The down-draft furnace appears to be one of the most successful appliances for smoke-prevention with smoky coals. As satisfactorily applied it involves the use of two sets of grate-bars, one over the other, so arranged that the draft passes downwards through the upper and lower sets of bars, or else passes downwards through the upper and upwards through the lower. Each set has its own fuel, but the intention is that the gases shall be distilled off from the fresh fuel on the upper grate, and shall be drawn downwards to mix with the hot products escaping from the lower where the solid carbon is burning. By this the temperature of ignition is maintained for the distilled gas, so that it shall burn with the abundant supply of warm air admitted for this purpose. Figs. 30 and 31 show boiler-settings of this type.

(6) The use of fire-brick or similar refractory material for the furnace or in the combustion-chamber (Fig. 32). This becomes hot by the impact of flame and gas, and keeps the temperature of the gas up to ignition. It imparts some of its heat to the boiler by radiation after it is once brought up to full heat.

(7) Preheating of the air-supply by hollow walls or flue-

FIG. 30.

Fig. 31.
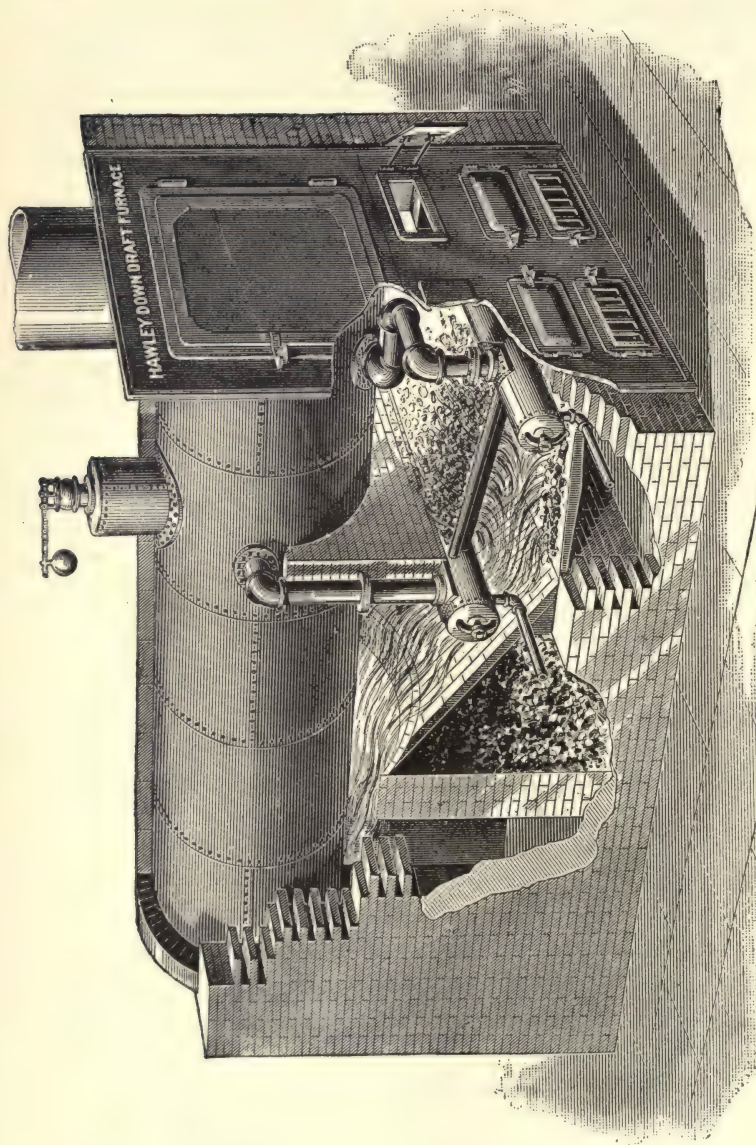
Fig. 3
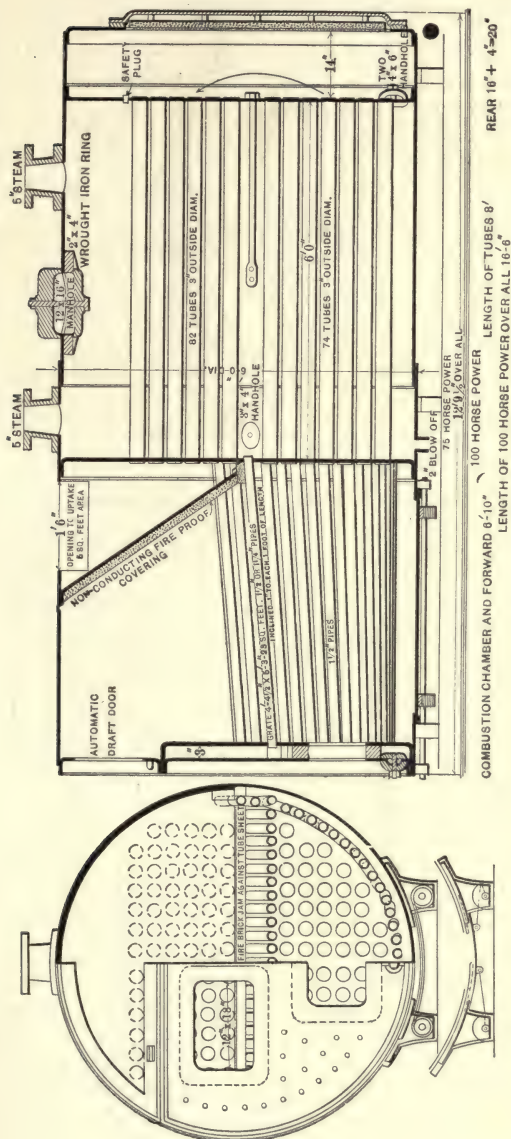
boxes which the hot gases surround while the fresh air flows within them.

The objections to most of the smoke-prevention devices have been that the introduction of such appliances diminishes either the economy or the capacity of the plant as compared with what it was when the chimneys were allowed to smoke. The excess of air diluting products of combustion explains a loss of economy and capacity, and the superior efficiency of the yellow flame, as compared with the colorless flame of perfect combustion, is also responsible in part for this result. The losses seem to be about 12 per cent of power or from 7 to 13 per cent of economy.

The term smoke-consumption or smoke-burning is an improper one. Lamp-black once made is incombustible and cannot be burned. The products of combustion are often colored brown by the presence of tarry or similar combustible matters, and these will ignite if the temperature be made hot enough. It is possible to prevent appearance of smoke by catching it in water through which the products of combustion pass, and in which the carbon is thrown down.

90. Mechanical Stoking.—Modern successful combustion is also much indebted to improvement in the grate-bar of the furnace for the securing of smokelessness. While the shaking-grate has made the fireman's work more easy, it is to the step-grate and the travelling-grate that smokeless combustion is more largely due.

In the step grate the bars are flat surfaces or treads arranged so that the upper one slightly overlaps the one below it, while leaving open for the passage of air the space which corresponds to the riser in stairway construction. It will be seen that this construction permits abundance of access of air with little or no possibility of coal dropping through the grate-surface; or the principle of a forced draft can be applied (Fig. 33). When the bars are laid across the furnace, as is usual, the slice-bar of the fireman can cleanse

each bar separately by working through the vertical opening
between the bars, or the method of firing may be used
whereby the coal is fed first on the upper bar, and from that
is gradually pushed down the steps from bar to bar until at
the bottom it will be pushed off with all available combusti-
ble matter utilized, and only refuse and ash remaining
(Fig. 33).

It is very easy to convert such a step-grate into a shak-
ing- or dumping-grate by arranging each bar so as to permit
a motion to tip its burden down the steps. This can be done
either by hand or mechanically.

The principle of successful passage of fuel from bar to bar
suggested in the previous paragraph leads to a construction
of grate which is known as the travelling-grate. The bars,
instead of being continuous and solid, are made up of a series
of short bars which are pinned together so as to form a flat
chain with the links edgewise. These chains, made endless,
mounted upon proper carrying-rollers at the front of the
furnace and at the rear, and having the width of the furnace-
area, can be driven by machinery attached to the rollers so
as to draw the chain from the front of the furnace to the
back, carrying on its surface the fuel to be burned. The
speed of driving should be so proportioned that the fresh fuel
charged at the front upon the travelling bed of the grate
should be completely burned during the period of its transi-
tion to the back, so that when a given chain of links reaches
the rear roller and is dropped over, there is carried with it
and dropped only the incombustible matter in that given
amount of coal. Such a grate is practically self-cleansing
and leads at once to the use of an automatic appliance for
feeding the fuel to it to make it complete. Fig. 34 shows
a typical travelling-grate, and the plan shown in Fig. 33
can be made automatic.

If the self-cleansing mechanical grate can be combined
with automatic or mechanical feeding of the fresh fuel which
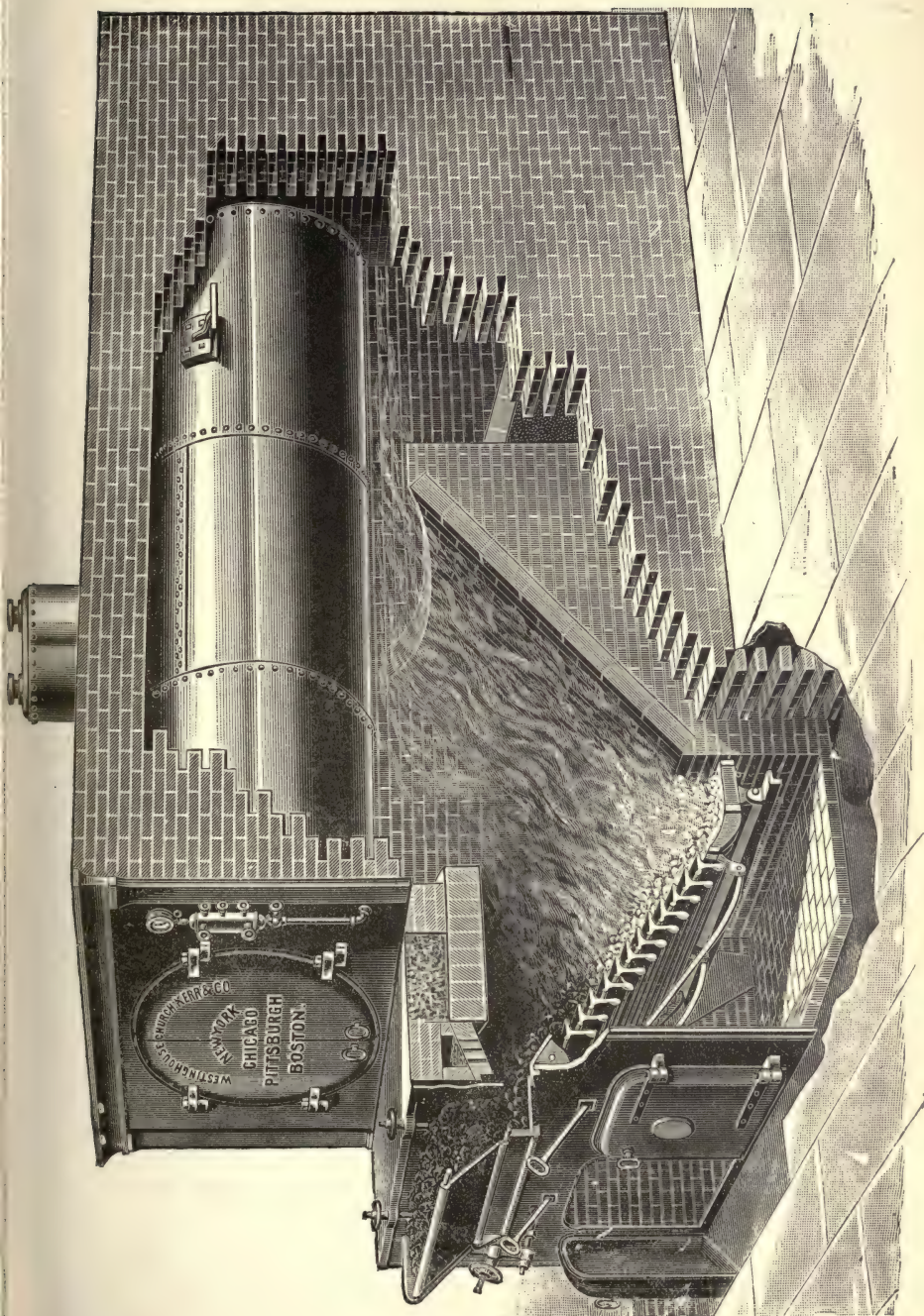
Fig. 33.

is to be burned upon it, it will be apparent that not only has the supply of fuel as a source of heat energy become continuous and uniform, but the combustion of the fuel is made regular and continuous because the fire is at all times in the same condition. Furthermore the labor of the fireman has changed from a hard muscular exertion of hand-firing to the skilled supervision of machinery of sufficient power to do the required work. In the mechanical stokers which have been

MECHANISM OF COXE'S CHAIN GRATE STOKER.
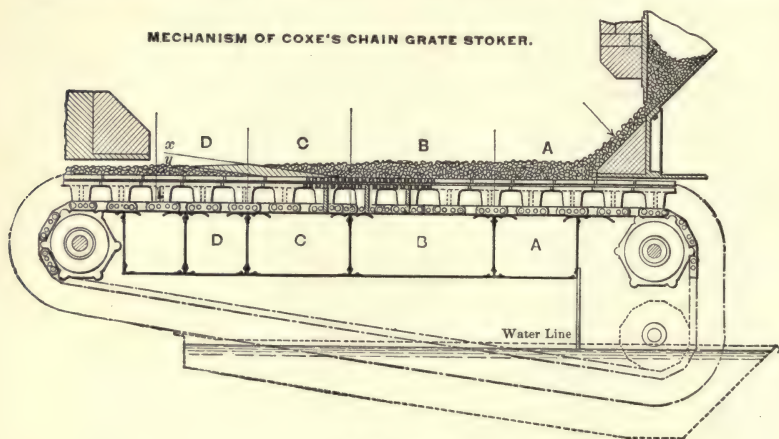
Fig. 34.

approved the coal is fed upon the travelling or mechanical moving grate from a hopper, either through an opening or between rails which carry ribs lengthwise so as to form pockets to receive the fuel. Thus the speed of these pockets measures the quantity of fuel delivered. The travelling-grate or the measuring-rollers can have their speed regulated by simple mechanical means connected with the steam-pressure; and if the air for combustion is supplied by mechanical means, the volume of that air can be regulated by the rise and fall of the pressure of steam by causing the latter to vary the speed of the engine which drives the fan or controls the valve which supplies the steam-jet. Figs. 33 and 35 show types

in which the motion of the step-bar itself causes the fuel to
be carried down the steps to be delivered as ash at the bot-
tom. It will be seen that the form of grate shown in Fig.
36 can also be very easily and properly fitted to the principle
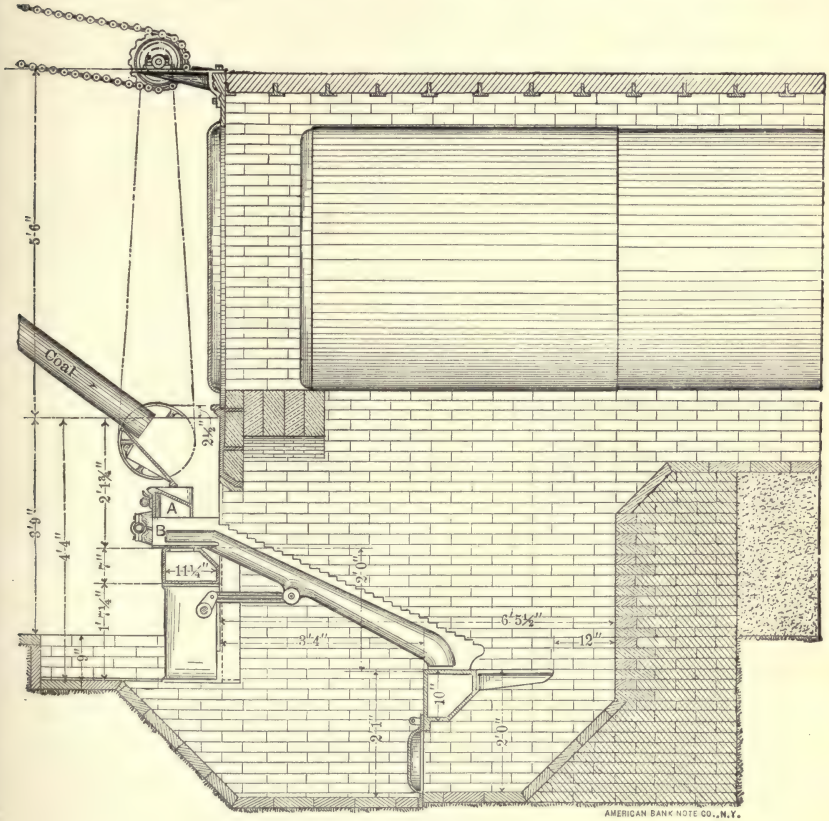
Fig. 35.

of automatic stoking. The supply of fuel to the hoppers at
the boiler-fronts will be done by the principle of mechanical
conveyors with elevators if the supply of coal in pockets can-
not conveniently be made overhead. If the coal-vault can
be over the boiler-room, the coal may descend by gravity

through proper spouts into the furnace-hopper without handling.

This principle of mechanical handling of fuel, combined with mechanical handling of ashes and with the principle of automatic control of the machinery of stoking as the steam-pressure may vary, gives to a modern power plant where the principle is applied all advantages derivable from doing away with human labor and replacing it with intelligent control of
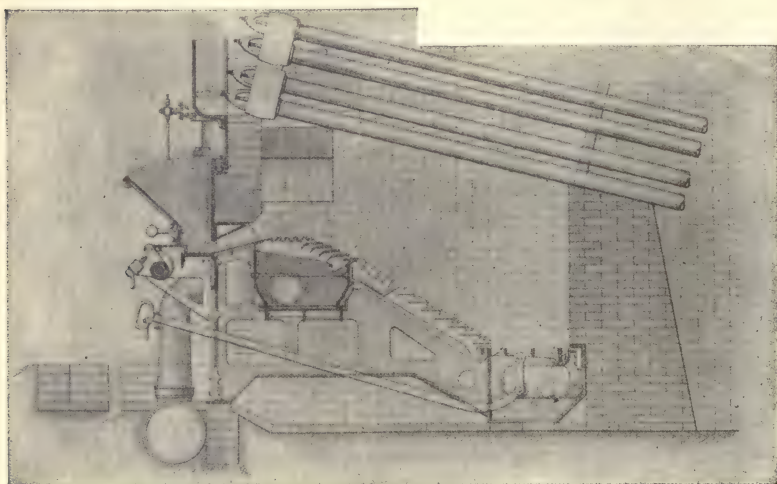
FIG. 36.

inanimate force. It has not been proved that the advantages from uniformity and continuous action always represent a surplus sufficient to pay for the increased cost of the installation, but the saving of labor expense usually leaves a margin, in a plant of any considerable size, which is abundant to offset such cost.

Mechanical stoking has not achieved its best success with the hard varieties of anthracite coal with which the fireman's labor is the least. Again, with certain varieties of bituminous coal which cake and melt it has been found that their working
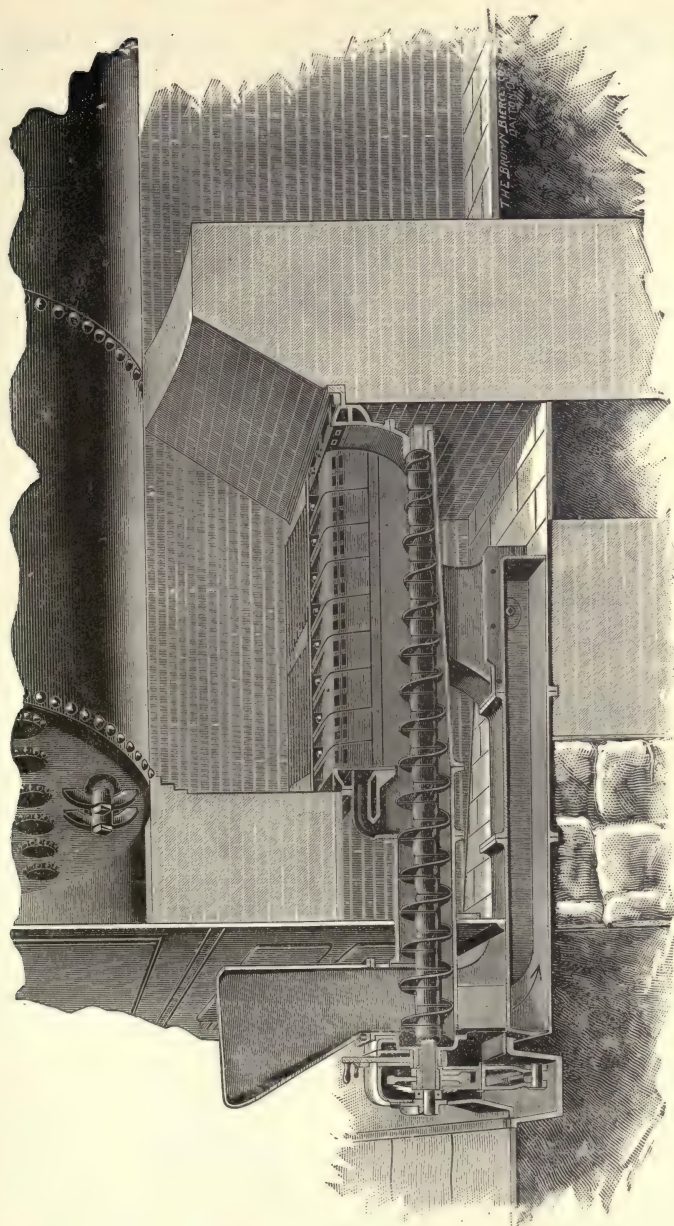
Fig. 37.

is not satisfactory in every case. Fig. 37 shows a form of stoker in which the feeding of fresh fuel is done from the bottom, so that the products of the first distillation are forced to pass up through the bed of incandescent fuel from which the gases have been removed. This brings them up to the point of ignition, and the slope of the sides of the bed of fuel is covered with coal in the condition of fixed carbon, which when completely burned falls off as clinker or ash at the sides of the grate, or is removed by slicing. Forced draft on the closed ash-pit or closed fire-room system can often be applied to this stoker with advantage (Fig. 38).

FIG. 38.

It has recently been suggested that a standard color scheme should be accepted by engineers and inspectors in dealing with the smoke problem. The Ringelmann scale is exhibited in Fig. 39, which proposes four standards. The smoke is to be observed against a clear sky, and its color
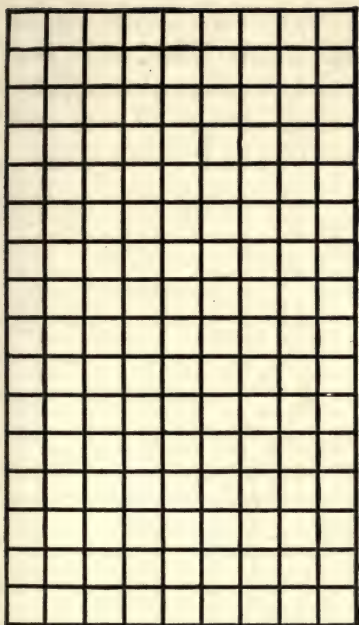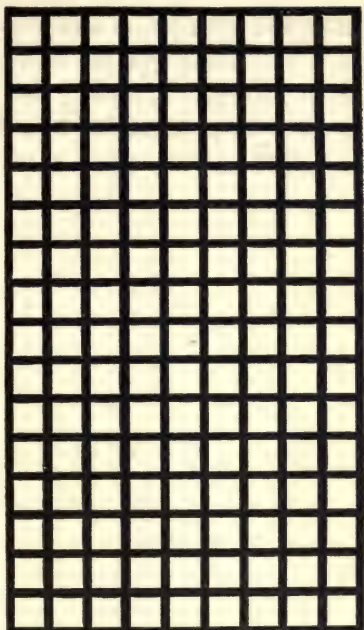
No. 1. No. 2.
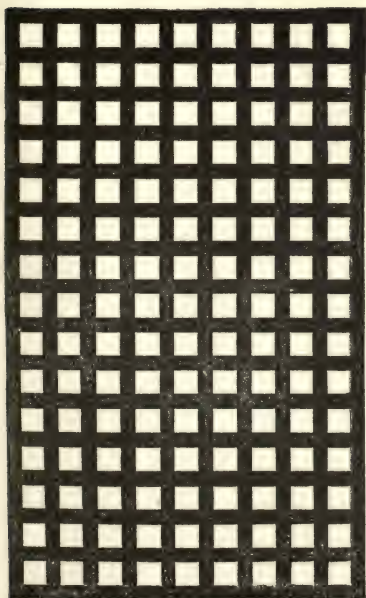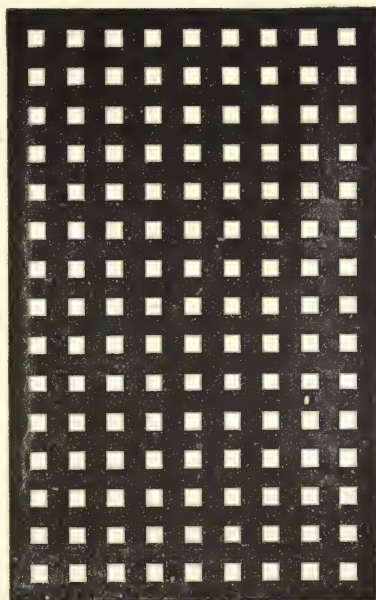
No. 3. No. 4.

THE RINGELMANN SCALE FOR GRADING THE DENSITY OF SMOKE.
Fig. 29.

compared with the effect upon the eye of an 8-inch square black-and-white grating of the scale standard held at 50 feet distance from the eye.   No. 1 would be pure white paper, and No. 6 in the series would be entirely black; hence each intermediate proportion corresponds to a 20 per cent range.

CHAPTER VIII.

TRANSFER OF HEAT. HEATING SURFACE.

91. Introductory.—In the four chapters which have pre-ceded (Chapters IV to VII) it has been the object to show how the energy stored in the fuel could be liberated there-from at the will of the engineer for motive power or indus-trial uses. The next step must be to examine how this liber-ated energy in the form of heat-units per pound of combus-tible can be made available for the doing of mechanical work. Two steps or stages are involved in this transformation: the first is the transfer of the heat to a medium convenient to carry the energy from the fire to the motor cylinder or organ of the machine; the second is the effect produced upon that medium by such increase of its previous heat energy, and a discussion of the availability and convenience of various media.

Certain necessary investigations are at once suggested:

1. How is heat transferred from one body to another.

2. What are the best media, or those which give most efficient transformations of the heat energy of the fire into motor energy.

It is also apparent that from this point onward the prop-erties of the medium used as a heat-carrier from the fire to the cylinder are likely to require to be taken into consider-ation either expressly or by implication, and that a sort of general division along this line seems to be required. This scheme of differentiation would separate heat-motors into

two great classes. The first class would include those in which the liberation of heat occurs directly in or around the motor-cylinder; the second class would be those in which the liberation of heat occurs in or at a second apparatus from which the energy passes to the motor-cylinder through pipes or passages by which generated pressure is supplied to the cylinder. The first class would include the gas-engines, the hot-air engines, the oil-engines, the explosive-vapor engines, the gunpowder-engines. The second class contains the steam-engines, the ammonia or volatile-vapor engines, the compressed-air engines, and other types in which a boiler or generator and a reservoir of pressure are necessary features in addition to or outside of the engine proper. The significance of this division will appear more manifest in later chapters and after detailed discussion.

Speaking more roughly and with less regard to exactness of detail, heat-motors may be classified into those using the permanent gases, which, like air, do not change their state under changes of heat-condition, which form one class; and those using water or other liquids which will form gases or vapors under increase of heat energy, which motors form the second class. This does not quite coincide with the fundamental idea of the previous division, and yet does not disagree with it and has some conveniences. The subject of transfer of heat, however, may be discussed with both classes in view.

92. **Transfer of Heat. General.**—In every heat-engine operating by pressure caused by heat, that pressure must be contained in or resisted by a closed vessel—usually metallic —and in most cases the heat is outside of this vessel and must be transferred to the motor medium within it. Hence the heat must be first transferred to the metal enveloping the medium, and secondly must pass from the metal to the medium, and thirdly must distribute itself through the medium if the latter has any extended volume. In oil- or gas-engines,

where combustion takes place directly in the working cylinder, only the two latter steps occur. The combined process of giving out heat by the fire, and its absorption by the medium, is called the transfer of heat. Refrigeration as a process is also a transfer of heat, but in the opposite direction, since the object is to diminish the heat energy of the body acted on, and not to increase it as in heating. In more exact language, the object of a process of transfer of heat is to increase the heat energy of the cooler body, and to equalize the intensity of heat-motion of their respective molecules. If the hotter body receives no increment of heat during the transfer process, it is refrigerated by the transfer. Ordinarily, of course, the heat condition of the hot body is kept as near uniform as possible, and heat energy passes constantly out to the absorbent body.

It will be at once apparent that the rate of transfer should be faster the further apart the heat condition of the two bodies, and that when their heat condition is nearly equalized the transfer per unit of time will be correspondingly diminished, and will become zero when both are in the same state of heat energy.

Experience and observation show that heat can be transferred by four processes:

(1) By radiation.
(2) By contact.
(3) By conduction in solids.
(4) By convection in fluids.

In radiation the two bodies are separated by a space. In contact they touch each other but are not one. In conduction the heat-motion is at first more active in one part of a solid body than in another, and that heat-motion is transmitted neither by radiation nor by contact with another body. In convection the cooler denser particles of a mobile fluid displace the hotter lighter particles,—which seems like a

transfer of heat, but is rather a mechanical displacement of particles until all are equally heated.

The principles of transfer by each method will be discussed hereafter.

When two bodies are equalizing their heat conditions by a transfer of heat from one to the other, as when a metal mass at one temperature is immersed in a fluid at another, the action may be expressed by an equation which shall express the gain or loss of heat in heat-units. It takes the following form:

$$U = w \times c \times (t_1 - t) = w' \times c' \times (t_2 - t).$$

In this U is the desired units of heat transferred; w and w' are the respective weights in pounds; c and c' are the respective specific heats; t_1 and t_2 their respective initial temperatures, and t their common final temperature. One body will have gained what the other has lost, and the value of U can be calculated from either. The application of this expression, however, must be restricted to the cases for which it is true.

93. Transfer of Heat by Radiation.—When a body is radiating heat, its condition is one in which lines of heat energy emanate from the body in every direction into space. Bodies which are in the path of these heat-lines receive the impact of the heat-waves, and their heat condition increases in intensity until their own tendency to transfer heat to other bodies precludes their rising any higher in the heat scale. Heat behaves like light in a transfer by radiation so far as the radiating body is concerned. Unlike light, however, the effect of heat on the absorbing body is cumulative, up to the point where the absorbent begins itself to transfer.

Heat from radiation appears to vary inversely as the square of the distance, because a body at a distance 2 from a centre of heat-motion receives only one quarter as many heat-impulses in a given time upon a given area as that same

area would receive at a distance unity; or the area at 2 to receive as many heat-impulses as are intercepted at 1 must have twice the height and twice the breadth needed at 1 (Fig. 40). If the rays also grew less intense by distance, the effect of distance would be to make the effect of heat
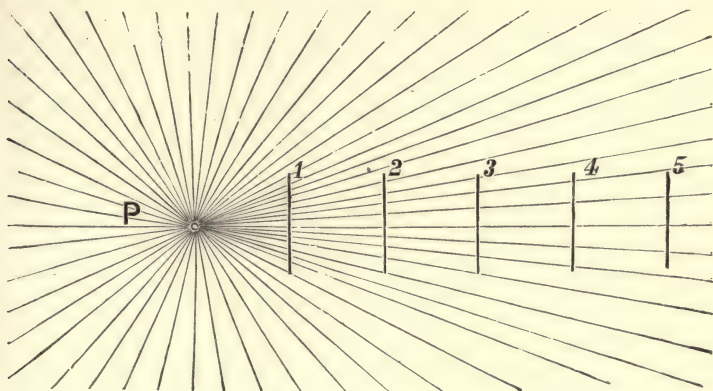
Fig. 40.

vary as the fourth power of the distance when absorbing areas were the same. This same area effect explains why radiation is less effective in heating inclined surfaces than when the heat-impulses are normal. The projected area is the effective one only.

Radiation has been studied by physicists with heated solids having a radiating area. Dulong and Petit's formula is, for metric units,

$$Q = C_1(a^{t_1} - 1).$$

C_1 is a constant depending on the surface of the radiating body, large for dark color and rough surface and least for smooth and polished surfaces light in color. a is the number 1.0077, a constant according to Dulong and Petit, variable according to De la Provostaye and Desains. Q is the quantity of heat in calories emitted from a unit of surface in

a unit of time. t_1 is the excess of the temperature of the radiant body over the absorbent in centigrade degrees.

The radiation formula is also written

$$Q = Ca^t(a^{t_1} - 1),$$

in which t is the temperature of the absorbent body.

Experimental formulæ by Hopkins give:

For glass, $Q = 9.566a^t(a^{t_1} - 1).$

For sandstone, $Q = 8.377a^t(a^{t_1} - 1).$

For polished limestone,

$$Q = 9.106a^t(a^{t_1} - 1).$$

In this, Q is the quantity of heat radiated per minute from one square foot of surface in units to raise a kilogram of water 1° C.

Better ideas about radiation are derived from tables of comparison. This table is by Leslie, the experiments being made at 180° F.:

Lampblack	100	Mica	80
Paper	98	Graphite	75
Resin	96	Tarnished lead	45
Sealing-wax	95	Mercury	20
Crown glass	90	Polished lead	19
India ink	88	Polished iron	15
Ice	85	Tin-plate	12
Red lead	80	Gold and silver	12

Darkness and roughness of surface increase radiation, while smoothness and polish diminish it.

Magnus' experiments at 270° F. give relations for radiating effect:

Blackened silver	100	Rock salt	13
Glass	64	Polished silver	9.7
	Fluor-spar	45.5	

The conditions in a boiler or a heat-engine furnace are very different from those of the foregoing, and the only data

usually referred to are from Péclet, who, having found the total heat of combustion, says that from a fire of coal, coke, or charcoal it is 50 per cent, from peat-charcoal 48 per cent, and from wood and peat about 25 to 29 per cent. Flame heats by radiation from the incandescent particles in it. Gases without such incandescent material seem to produce no effect by radiation however high their temperature.

Radiant heat does not warm air or other gases directly. Open fires warm only the objects and persons in a room, which in turn warm the air by contact.

94. Transfer of Heat by Contact.—By far the most important in transfer of heat in the heating of buildings, and in the heating and cooling of air in engines, is the interchange when the two bodies are in contact, as when the hot products of combustion pass over the metal of the boiler and give up their heat to it. Transfer by contact is also of primary importance in refrigeration.

For contact of solids with fluids the Péclet formula is

$$Q = C_1 t^{1.233},$$

the notation as before. Balfour Stewart gives as Hopkins's formulæ:

For air, $\qquad Q = 0.0372 \left(\dfrac{p}{720}\right)^{.45} t_2^{1.233};$

For CO_2, $\qquad Q = 0.0359 \left(\dfrac{p}{720}\right)^{.517} t_2^{1.233}.$

when p is the pressure of the gas in millimeters, and Q is the quantity of heat emitted from one square foot as in § 93.

These data also are not of great significance for design of generators, while more practical than the radiation results; but the hot gases do not heat the motor fluid directly, but heat the metal of the enveloping reservoir, which conducts the heat to the motor-fluid.

It is interesting to note, however, that in these two formulæ for radiation and contact the difference in temperature enters in one as an exponent, and in the other as a coefficient. Hence for a given difference in temperature radiation will be enormously more effective than contact in transferring heat. This can be shown graphically by the diagram Fig. 41, where differences of temperature are abscissas and
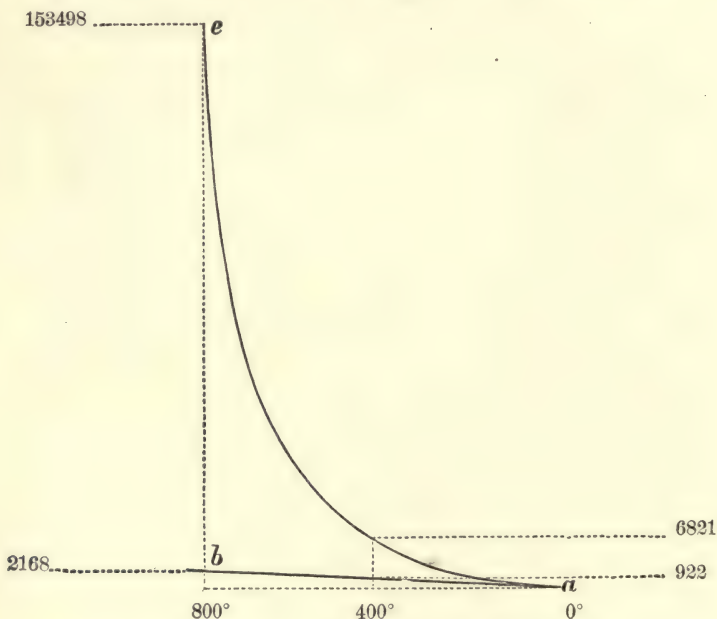
FIG. 41.

heat units, Q transferred are ordinates. The curve ae is for radiation, the line ab for contact.

The transfer for 800° difference of temperature is over 70 times as great by radiation as by contact. This is one of the reasons for the superiority of flaming coals over short-flame fuels, and for the lessened economy of gas-firing from gas made from coal in a separate generator. Anthracite as a short-flame fuel requires a large furnace area, as its heat is

mainly radiated from the solid carbon and not from the flame.

Rankine's formula for transfer by contact per square foot of boiler heating-surface per hour is

$$Q = \left(\frac{T' - T}{a}\right)^2,$$

in which T' and T are the temperatures of the two fluids in contact with the two faces of the metal, and a is a factor varying from 160 to 200. This he calls a rough approximation, and Q is B.T.U. and differs from the Dulong and Petit formula in substituting 2 for 1.233 and changing the constant.

In the transfer of heat in the heating of air by coils or radiators, using contact of air and hot metal in the rooms to be heated, or in the direct-radiation system, it is usual to calculate the condensation representing the transfer at 2 to 3 heat-units per square foot of transferring surface per degree difference in temperature, per hour. That is, experiments show such condensation to average from 1.66 heat-units with ineffective radiators up to 2.25 with flat pipe-coils or good radiators. Otherwise stated, it appears that with low-pressure steam of one or two pounds pressure the condensation per square foot ranges from 0.25 to 0.30 pound per square foot per hour. In the transfer of heat to air from coils in ventilating systems, where the air from out of doors is passed over metal surfaces to warm it before distribution, the rough rule may be used that one square foot of radiating-surface with steam at 212° will heat 100 cubic feet of air from zero to 150° per hour or 300 cubic feet from zero to 100° in the same time; or the relation of specific heats for equal masses may be used to calculate the water required. If the specific heat of air be called 0.238, while the specific heat of water is called unity, it is obvious that one pound of water will heat

$\dfrac{1}{0.238}$ = 4.2 pounds of air through a given range. If a
pound of air occupies 12.39 cubic feet, the lowering of one
pound of water one degree will raise $4.2 \times 12.39 = 52$ cubic
feet of air one degree.   If low-pressure steam is used, so that
966 units are transferred in cooling steam to water (§ 58) at
212°—which may be called 1000 units without sensible error
—it appears that the pounds of water to be made into steam
per hour bear to the cubic feet of air required to be warmed
one degree per hour the relation of unity to 50,000.   Roughly
speaking, the evaporation of 30 pounds of water per hour
will heat a cubic capacity as follows:

Brick dwellings, in blocks, as in cities.....	15,000 to 20,000	cu. ft.
" stores, " "	10,000 " 15,000	"
" dwellings, exposed all round........	10,000 " 15,000	"
" mills, shops, factories, etc...........	7,000 " 10,000	"
Wooden dwellings, exposed............	7,000 " 10,000	"
Foundries and wooden shops..............	6,000 " 10,000	"
Exhibition buildings, largely glass, etc....	4,000 " 15,000	"

The water to be evaporated per hour having been found,
the calculation of boiler, grates, and the like will be made by
methods to be treated in later chapters.

In a negative transfer where cold brine is circulated in
coils to cool a room (see Chapter XXII), each square foot of
surface can take care of 1200–1300 heat-units per hour, to be
withdrawn from the material to be cooled.

95. Transfer of Heat by Conduction.—When one end of
a bar of metal is exposed to heat by putting it into or near a
fire, the heat energy imparted at the one end is conducted to
the parts farther from the source of heat by overcoming a
certain resistance to such increase of heat energy.   This
resistance has been called the thermal resistance; or the con-
ductivity of the metal is the reciprocal of such resistance.
The transfer is cumulative, since the bar grows hotter and
hotter, up to the point at which the transfer of heat away
from the bar by radiation or contact or both becomes equal

to the amount which it receives by conduction in the same time.

The conductivity of metal and the rate of such conduction are primary elements of transfer of heat in all cases where the medium to be heated is enclosed in a vessel upon whose exterior surface the heat of the fire is brought in order to raise the temperature of such enclosed medium. The outer layer of the metal nearest the fire should be as nearly at the temperature of the fire or the hot gases resulting from combustion as the efficiency of the transfer of heat by radiation or by contact will permit, provided there were no conduction to the presumably cooler inner layers. The more instantaneous the conduction, and the less loss of heat energy in the process, the more nearly will the inner layer which touches the enclosed medium approach to the temperature of that which heats the outer layers. When the conducting metal is thin—having but a small fraction of an inch in thickness—the transfer is practically complete, and with a medium enclosed such as water and having a high specific heat (§ 12) the metal has throughout the same temperature, which is that of the cooler fluid. With thicker walls of metal, the greater mass to be affected by changes of heat energy, or through which the thermal resistance may act, will increase the difference of temperature between the outer hot layer and the inner cool layer. In other words, for a given transfer per unit of time the outer layer must be hotter with thick plates than with thin. Thin boiler-plates absorb heat more effectively from the hot gases which pass over them because the outer layer is further removed from the temperature of such gases than when the plate is thick, particularly when the gases are moving rapidly and the time for absorbing heat from each pound of gas is short.

Conduction is expressed by a formula

$$Q = \frac{C(T' - T)}{e}$$

in which e is the space separating the two surfaces which are at the temperatures T' and T respectively, and C is a coefficient for each material to be determined by experiment, and the conductivity to be relative to a standard.

Relative conductivities as measured by Wiedemann and Franz are:

	In Air.	In Vacuo.
Silver	100	100
Copper........................	73.6	74.8
Gold..........................	53.2	54.8
Brass	23.6	24
Tin...........................	14.5	15.4
Iron..........................	11.9	10.1
Steel	11.6	10.3
Lead.........................	8.5	7.9
Platinum......................	8.4	7.4
Palladium	6.3	7.3
Bismuth......................	1.8	...

Conductivity of metals drops as the temperature increases. For iron, with an increase of 100° F. the foregoing figure diminishes 15 to 25 per cent (Forbes).

The quantities of heat in B.T.U. transmitted per second through an area of one square foot one millimeter in thickness for one degree F. difference in temperature are approximately for the following materials (Neumann):

Copper................	41.2	Iron	6.1
Zinc..................	11.4	German silver..........	4.1
Brass	11.2	Lead	1.4

The absolute values of conductivity in liquids are uncertain and are certainly low. The received values (Guthrie, Philos. Trans., 1869) have been determined in terms of the resistance to transfer of heat, which is the reciprocal of the

conductivity and is called the thermal resistance. This thermal resistance is for

Water................	1	Sperm-oil............	8.85
Glycerine.............	3.84	Alcohol..............	9.09
Acetic acid............	8.38	Turpentine..........	11.75

The conductivity of liquids is greater at higher temperatures than at low; but when convection is not possible it is small in any case.

Despretz puts that of water $= \frac{1}{100}$ that of copper.

Conducting-power of gases is very slight, and it is often supposed they have none. Magnus gives that for air $= \frac{1}{1400}$ that of lead.

In a boiler, when the hot gases are at one constant temperature and the water at another, it would appear that the thickness of the plate would not affect the rapidity of transfer. But the gases are moving in the apparatus at speed, and do not stay long in contact with the plate; hence a thick wall prevents the heat of gas from being so efficiently abstracted by retarding the equalization downward to the temperature of the cooler (the water), and so heat is wasted by escaping unreduced. This loss should be diminished by proper use of what are called "retarders."

96. **Transfer of Heat by Convection. Circulation.—** The process whereby heat is transferred from the outer layers of a fluid to the inner ones, or from the bottom to the top, must differ from the simple conduction which takes place in a solid. The molecules being easily mobile among themselves, the cooler ones being heavier tend to descend and displace the lighter and warmer ones within the confining vessel, and there is thus produced a continual movement of the confined medium, whereby imparted heat is carried about within it, the hotter part going to the top and the cooler to the bottom. This movement due to differences of specific gravity caused by heat is called the convection of heat. It

is much less rapid than the process of conduction in solid matter, but is the only way in which large bulk of water or gas can be heated. The source of heat should obviously always be on the bottom of such masses of fluid. When water is being heated and vaporized as in a steam-boiler there occurs not only the convection process, but as soon as steam-gas bubbles begin to form, which are much lighter than the water, a different movement begins, accelerated in character as compared with the earlier convection, because of the difference in weight of the same bulk of steam and of water. The steam-bubbles tend to rise to the surface, and tend to accelerate the convection when they conform to its direction, and to disturb it when opposed to its direction. This motion in a steam-boiler is called the circulation, and is of primary importance as respects the transfer of heat. It can be directed but cannot be antagonized. Surprising results have been secured by mechanical circulation, where the speed of motion is greater than it would be if differences of specific gravity were alone depended upon.

The difficulty of transfer of heat from gases or to them makes it necessary that the gases should be finely divided into thin layers or small bulks if the transfer of heating or cooling effect must be rapid. This principle underlies the use of small tubes in tubular boilers, and is a sound one if only abstraction of heat from hot carbonic acid is the object of such tubes. Small tubes are not favorable to combustion, and will make a gaseous fuel a smoky one. The forcing of hot gases in large flues to move in eddies by baffle-plates or cross-partitions causes a continual convection motion which is favorable to the abstraction of heat by cooling surfaces presented to the gases.

97. General Remarks on the Transfer of Heat.—In discussing the usual formula for an exchange of heat,

$$Q = C_1 \times w \times (t' - t),$$

which may be written

$$Q = C_1 \times v \times D \times (t' - t),$$

in which Q = quantity of heat transferred;

w = weight of gas passing per unit of time;

v = its volume in cubic feet, and

D = its weight per cubic foot;

$t' - t$ = the range of temperature change.

It will appear—

1st. The quantity will increase as the difference in temperatures. Hence circulation is beneficial, and the hotter gas should meet the hottest water, and the coolest gas the coldest water, to keep the difference a maximum.

2d. The denser the gas the more heat it transfers. Hence plenum and forced-draft systems are more efficient than aspiration systems.

3d. Liquids transfer heat faster and more efficiently than gases by reason of greater density. The presence of moisture in an air causes it to take more heat from the body than a dry air. Hence the coldness of damp winter days. The cylinder-walls give heat to the damp air of the exhaust of the steam-engine at a rate seventy times faster than if that gas were dry and free from moisture.

98. Heating-surface.—The area of metal exposed on one side to the heating effort of the fire and on the other to the medium to be heated by conduction of heat through that surface will be called the heating-surface. The cooling-surface is the same thing, only with the direction of transfer reversed. The practical result therefore to be sought by the engineer and designer is the proportioning of the absorbing surface for heat so that with a given liberation of heat-units in the fire there may be a transfer of heat energy to the working medium with the containing vessel which shall raise its heat energy to the greatest possible extent. With the steam-

boiler this means the evaporation of a given weight of water into steam at a given pressure with the combustion of a given weight of fuel or combustible, per unit of time. Confining the discussion for the present to the steam-boiler, it may be said that the quantity of heat transferred to the water will therefore depend upon the extent or weight of that contact-surface of metal, and the difference of temperature between that metal (or the water which touches it) and the imparting source of heat—fuel, flame, or gas. Hence the amount of heating-surface for a given evaporation of water or absorption of heat will be fixed—

1st. With relation to the rate of combustion to be employed—since the faster this rate the higher the temperature of the fire and the gases.

2d. In some relation to the absolute quantity of heat supplied in a given time—which is a relation to the square feet of grate-surface on which the fuel is burned, if the rate of combustion is assumed.

It will appear at once that the terminal temperature of the gases when leaving the generating apparatus must be considered and fixed. If the gases are too cool, they do not transfer heat to the heating-surface and water; chimney-draft is dependent on a certain minimum temperature in the stack as compared with the outer air. If the gases are too hot, heat is wasted in the chimney, because there was not surface enough to abstract the heat as fully as might be, and coal has been burned to waste, heating the stack and outer air and not the water.

Hence it is usual to fix the terminal temperature at about 600° F. (the maximum draft temperature, § 80), since steam at 250 pounds pressure has a temperature of 401° F., and these gases will give off their heat to the steam even when the difference is reduced to 200°. For lower pressures they are so much more effective, and can be cooler if the draft need not be considered.

If the terminal temperature in the flue Ob (Fig. 42) be set at 600°, represented by an ordinate bf, and the curve for the transfer of heat be drawn through f according to the formula $Q = Cw(t' - t)$, there will be found a point A which will indicate the initial difference of temperature between the fire
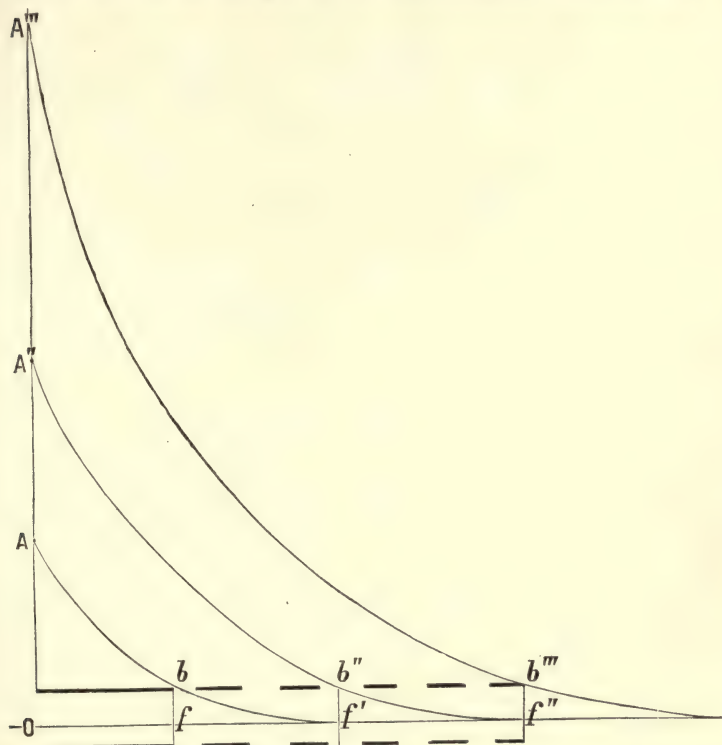
FIG. 42.

and the water such that the heat would be abstracted down to 600° in that extent of contact and transfer.

But if a higher rate of combustion be assumed, and a higher initial temperature represented by A'', then the curve will not pass through f as before, but through a point f' beyond it, so that with an extent of heating-surface represented

by the length Ob, the gases would leave unduly hot, and the heating-surface should be extended till b'' is reached, at which point the terminal temperature is the same 600° as for the first case. For a higher difference the third curve results, and an extent of heating-surface as much greater than the first case as Ob''' is longer than Ob.

This increase of heating-surface to absorb greater amounts of heat is not usually done by lengthening the boiler as indicated, but by increasing the number of tubes, and the diameter of shell as well as the length.

Experimental data on this subject have been obtained in two ways: by keeping the heating-surface constant and varying the rate of combustion, or by finding the increase of heating-surface to keep an evaporation constant. The following table is by Isherwood, for a marine tubular boiler using anthracite fuel and having a constant heating-surface 25 times the area of the grate.

Pounds of coal per hour per square foot of grate	6	8	10	12	14	16	18	20	22	24
Pounds of water evaporated from and at 212° per lb. of coal....................	10.5	10.4	10.1	9.5	8.9	8.2	7.7	7.3	7.0	6.8

The following table from D. K. Clark, "Railway Machinery," gives for higher rates of evaporation the relation between heating- and grate-surface to maintain a constant evaporation of 9 pounds of water for each pound of coke:

Pounds of coke per square foot grate	14	19	25	31	38	47	56	65	76	87	98	110	125	129	153
Heating - surface per square foot grate....	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100

Hence it appears necessary to decide upon the conditions of rate of combustion to be anticipated as usual, and fix the heating-surface accordingly.

The following table gives accepted data on this subject:

Type of Boiler.	Pounds of Coal per Hour per Square Foot of Grate.		Draft.	Authority.
	Anthracite.	Bituminous.		
Land................	6–12	12–27	Chimney	Whitham
"	....	65–80	Forced	Seaton
Marine	7–16	12–27	Chimney	Shock
"	....	18–20	"	"
"	....	20–30	Forced	Seaton
Cornish	....	4	Chimney	Rankine
Ordinary flue........	....	10	"	"
" factory.....	....	12–16	"	"
" marine.....	....	16–24	"	"
" locomotive.	....	40–120	Forced	"
Average...........	....	12–18	Chimney	Rankine

Thurston's formula for the rate of combustion per square foot of grate with a given chimney is:

$$\text{Rate} = 2\sqrt{\text{height in feet}} - 1.$$

99. Ratio of Heating-surface to Grate-surface.—There is further a certain agreement that the ratio between the grate-area and the heating-surface shall be generally maintained, approximately as follows:

With a grate-area of unity the heating-surface will be:

In Plain cylinder boilers........ .... 10 to 15, average 12
     Cornish flue-boilers................. 30 " 40, " 35
     Elephant boilers.................... 25 " 40, " 33
     Flue-boilers....................... 17 " 25, " 21
     Tubular boilers.................... 25 " 30, " 28
     Traction-engine boilers............. " 32
     Marine (Martin type) boilers .... " 25
     Locomotive-boilers................ 40 " 100, " 75

The Morin and Tresca rule, which has influenced much European practice, was to fix for each type a rate of combustion per square foot of heating-surface which was not to be

exceeded, and is given in the second column of the following table. Then the ratio and rates were as in the other columns.

Type of Boiler.	Pounds of Fuel per Sq. Ft. H. S.	Ratio H. S. to G. S.	Combustion per Sq. Ft. Grate.
(*a*) Land....................	0.6	24 : 1	14.4
(*b*) Marine................	0.5	28 : 1	14
(*c*) Locomotive............	0.8	70 : 1	56

Old English practice was to require and expect an evaporation of one cubic foot of water (62.4 pounds) per square foot of heating-surface when water was delivered at 60° F. and evaporated at 212°.

Old U. S. Navy practice was to allow 8 pounds of anthracite coal to the cubic foot per hour to raise it from 212° to steam at 30 pounds pressure. This required ⅔ of a square foot of grate (at 12 pounds per square foot per hour), and with a ratio of 25 to 1 the heating-surface for this unit was 16⅔ square feet.

From Isherwood's historic experiments the following table is taken:

Type of Boiler.	Heating-surface per I. H. P.	Water per Pound Combustible.	Combustible per Hour per Sq. Ft. Grate.
Marine tubular...........	19	12	11
" "	19	10.5	10.5
" "	16	11.2	9.16
" "	16.8	11.8	9.3
Vertical water-tube.......	20	12.4	10
Horizontal flue...........	15.6	11.2	9.9
" "	15	10.4	9.3
Marine tube.............	18	11.1	11.2
			Average 10.2

Another series of data is the following:

Type of Boiler.	Square Feet of H. S. per H. P.	Coal per Hour per Sq. Ft. H. S.	Relative Economy.	Relative Steaming Rapidity.	Authority.
Water-tube..........	10–12	.3	1.00	1.00	Isherwood
Tubular.............	14–18	.25	.91	.50	"
Flue................	8–12	.4	.79	.25	Trowbridge
Plain cylinder........	6–10	.5	.69	.20	"
Locomotive..........	12–16	.275	.85	.55	"
Vertical tubular......	15–20	.25	.80	.60	"

100. Evaporation in Boilers per Pound of Coal.—
Again, the design of the boiler may be approached directly
on the basis of water to be evaporated per pound of coal
burned.

It will be recalled (§ 22) that a pound of carbon of calo-
rific power 14,400 will evaporate as many pounds of water at
212° into steam at 212° as 966 (which is the number of heat-
units required to do such evaporation) is contained in 14,400.

$$\frac{14400}{966} = 15 + \text{lbs.}$$

If commercial coal is used instead of pure carbon, as stand-
ard, having a calorific power of 12,000 by reason of ash and
moisture in it, the pounds of water per pound of coal will be

$$\frac{12000}{966} = 12 + \text{lbs.}$$

It is doing very well in a test when 11 pounds is reached, and
in every-day service 7, 8, and 9 are acceptable. Less than
7 is poor.

If then the pounds of water per hour required in the form
of steam are known, the grate-area will be determined, and
the ratio of heating-surface to grate-area is taken from the
foregoing tables.

The weight of water and steam for an engine service can

be found from volume and pressure at cut-off or at release from an indicator-card by consulting steam-tables for the weight per cubic foot at those pressures.

Or if no tables are available, it can roughly be calculated as follows: Volume of cylinder in cubic feet $\times 2Rpm \times 60$ = cubic feet of steam per hour at boiler-pressure and no cut-off. This volume in cubic feet multiplied by

$$\frac{1}{n} \times \frac{p}{1} \times \frac{62.5}{1} \times \frac{1}{1700} = \text{lbs. water per hour,}$$

when n = point of cut-off in terms of piston-stroke;

p = pressure in atmospheres at point of cut-off;

62.5 = pounds per cubic foot of water;

1700 = multiplier to reduce steam at atmospheric pressure to water at same pressure, since 1 cubic inch of water makes 1700 cubic inches of steam at one atmosphere pressure.

101. Water per Horse-power per Hour.—Or again, experiment has shown that in various grades of engine an engine horse-power should be developed with the following pounds of water:

High-grade compound 16–20
Condensing single 22–24
Good large non-condensing..................... 28
Average size condensing....................... 30
Small 30–45
Pumps, elevators, and non-expansive engines   50 upwards.

The American Society of Mechanical Engineers, choosing a safe figure, has said a boiler of N horse-power should evaporate $30N$ pounds of water from a feed-water temperature of 100° to steam at 70 pounds pressure. This is $34\frac{1}{2}$ pounds of water (34.488) evaporated from and at 212° with easy firing, moderate draft, and ordinary fuel, and showing good economy. By forcing the boiler should be able to do one third more.

This standard boiler horse-power is equivalent to a development of 33,305 thermal units ($34.488 \times 965.7 = 33.305$) and therefore corresponds to a condition of economy belonging to a combustion of $\frac{33305}{12000} = 2.8$ pounds of coal (usually called 3) per horse-power per hour in the engine. Hence from these data also a boiler-, grate-, and heating-surface can be derived. If this principle be applied to the data in § 99, the accepted proportion of $\frac{1}{8}$ of a square foot of grate per horse-power is seen to be the result.

Still another path to solution is the assumption of $11\frac{1}{2}$ to 12 square feet of heating-surface to the horse-power on the above unit; or about 3 pounds of water will be evaporated from and at $212°$ per square foot of heating-surface. At slow rates of evaporation more square feet will be required. Kent gives the following table (p. 678):

Pounds H_2O from and at 212° per square foot heating-surface per hour.............	2	2.5	3	3.5	4	5	6	7	8	9	10
Square feet heating-surface per H. P.. ...	17.3	13.8	11.5	9.8	8.6	6.8	5.8	4.9	4.3	3.8	3.5
Ratio H. S. to G. S. at ⅛ sq. ft. of G. S. to 1 H. P....	52	41.4	34.5	29.4	25.8	20.4	17.4	13.7	12.9	11.4	10.5
Probable relative economy............	100	100	100	95	90	85	80	75	70	6⁻	60
Probable temperature of chimney-gas......	450	450	450	518	685	652	720	787	855	922	990

As two examples the following cases will illustrate the different results from assuming different conditions:

No. I. Chimney-draft: 12 pounds of coal per square foot grate per hour evaporation = 9 pounds of water per pound of coal. Required to evaporate 5000 pounds of water per hour.

$$\frac{5000}{9 \times 12} = 48 \text{ square feet of grate.}$$

If H.S. : G.S. :: 25 : 1, then heating-surface = 1200 square feet.

No. II. Artificial draft: 60 pounds of coal per hour per

square foot grate; 7 pounds water per pound of coal. Then

$$\frac{5000}{7 \times 60} = 12 \text{ square feet of grate.}$$

And if H.S.: G.S. :: 70 : 1, then heating-surface = 840 square feet.

Case I would probably be two boilers of 24 square feet each.

$$2 \times 24 \times 12 = 576 \text{ lbs. coal per hour, and}$$
$$576 \times 9 = 5184 \text{ lbs. water per hour.}$$

Case II. $12 \times 60 = 720$ lbs. coal per hour, and
$$7 \times 720 = 5040 \text{ lbs. water per hour.}$$

This illustrates an advantage of central power stations over road generation of steam for short lines.

102. Refrigerating-surface.—The transfer of cold in refrigeration and condensing is the same in principle as the reverse transfer of heat.

The standard experiments are Joule's (*Jour. Franklin Inst.* 1862) and Isherwood's (Shook's "Steam-boilers," p. 58).

Isherwood's results are:

1. The number of heat-units per hour transmitted per square foot of surface is in direct ratio to the difference in temperature of the sides of the intervening metal.

2. Within limits, the rate of transmission of heat through a metal is independent of its thickness ($\frac{1}{8}$, $\frac{1}{4}$, $\frac{3}{8}$).

3. The thermal conductivities of four metals is as follows:

Material.	Thermal Conductivity or Heat-units per Hour per Square Foot for 1° Difference Fahrenheit.	Relative Thermal Conductivity.
Copper (refined).................	642.543	1
Brass (60 Cu, 40 Zn).............	556.832	0.866607
Wrought iron....................	373.625	0.581478
Cast iron (several times remelted)..	315.741	0.491393

Prof. Jay M. Whitham's formula for condensing-surface is

$$S = \frac{WL}{ck(T_1 - t)}.$$

In this, S = condensing-surface in square feet;

$\quad L$ = latent heat of saturated steam at temperature T_1;

$\quad T_1$ = temperature of steam at pressure recorded by the vacuum-gauge;

$\quad t$ = mean temperature of condensing water—the half-sum of initial and final temperatures;

$\quad k$ = perfect conductivity of 1 square foot of condensing-surface of metal used from above table;

$\quad c$ = fraction denoting efficiency of condensing-surface: probably about .3.

Standard experiments give a value for ck = 180, hence

$$S = \frac{WL}{180(T_1 - t)}.$$

Joule found the resistance to conductivity to be due to a film of water on each surface, and that to circulate the condensing water rapidly was to increase the conductivity of the metal.

In cooling brine by coils containing a cold fluid, experimental values give a result of 79 square feet of coil-surface to dispose of 100,000 heat-units negative per hour.

103. Conclusion.—While further topics belonging to the proper appliances for the actual liberation and transfer of heat might properly be introduced here, to do so would make the discussion too voluminous to be convenient. Students will find the subjects of boiler-setting, boiler accessories, care and management of boilers, and the like fully treated elsewhere, to which references will be found in the Appendix.

CHAPTER IX.

MEDIA USED TO TRANSFER HEAT ENERGY.

105. Introductory.—A heat-engine has been heretofore defined (§ 6) as one in which an effort in pounds was exerted through a space or path in feet, and where such effort was the result of a pressure on an area, such pressure being caused by heat. The preceding chapters have been concerned first with the generation or liberation of heat from the storage of such heat energy in fuel; and secondly, with its transfer to a proper medium to act upon the piston which is the mechanical organ to receive that effort. The subject next to be entered upon must then be the effect of increase of heat energy in proper motor media, and the laws of their action under changes which they may undergo in the amount of heat represented in heat-units on the absolute temperature scale.

The properties of these media and the effects of heat changes upon them are physical phenomena for investigation by the physicist in his laboratory. The engineer, however, is concerned with a comparatively narrow range of those phenomena and properties which are properly within the domain of physics in the field of heat.

106. Solids, Liquids, and Gases.—It will be generally agreed that the matter of the earth (and, so far as known, of the universe) appears in solid or in fluid form. The solid matter is that which can be changed in shape or figure only by considerable exercise of force, while the particles

of the fluids are mobile among each other, and only remain in any relation to each other by the exertion of some force. The fluid matter is again subdivided into liquids, which have a considerable weight per cubic foot and some cohesion of particles and no inherent elasticity or tension at usual pressures; and gases, whose weight per cubic foot is small, and which have at all usual pressures a tendency to expand and fill larger volumes as soon as such increased volume exists in connection with the gas. Instead of the solid and fluid subdivision, that into solids, liquids, and gases is preferred by many, and for present purposes is most convenient.

This threefold division is further of advantage since the accidental conditions of temperature and pressure may cause the same matter to appear in one or the other of the three states. Water, for instance, at atmospheric pressure is a solid below 32° F., a liquid between 32° and 212°, and a gas above the latter point. Mercury is a solid below 39° below zero Fahrenheit, a liquid up to 648° F., and a gas above this point. All of the usual metals are solid as they commonly are found, will become liquid or melt at a sufficient temperature, and are volatilized at the temperatures of the electric arc or furnace and in that of the sun. On the other hand, many of the substances which at atmospheric pressure and ordinary temperatures are known as gases will become liquid by sufficient pressure and lowering of their temperature. Such condensable gases are ammonia, sulphurous acid, some petroleum products, carbonic acid gas, the air, and others. When the pressure is released or the temperature is raised, they will return to the condition of gases. It is proper to say, therefore, since the foot-pounds required to compress a gas can be translated into heat-units by multiplying by 778 (§ 10), that the state of a body as to its condition as a solid, a liquid, or a gas is dependent upon its heat condition. A gas which has not yet been made into a liquid by pressure or cold or both is called a permanent gas. Improvements in apparatus, how-

ever, are continually shifting gases from the list of permanent gases into that of condensable gases. A more exact definition will be given in a following paragraph.

107. General Characteristics of a Medium to be used in a Heat-engine.—It has been already established (Chapter II, § 7) that the effort of a piston-engine in foot-pounds can be expressed by the product of the two factors PV, either per stroke or per minute or per pound. It is the effort of the engineer who is to use the motor energy of heat to make this product as large as possible with the least expenditure of his store of heat in the fuel. With a cylinder which has been actually constructed in cast iron or other material the value for V has become a fixed quantity in any one engine, so that P must be the quantity which it is desired to have increase as more heat energy is supplied. Hence the medium to be used in the heat-engine should be one in which the following equation should be true:

$$PV = ZT,$$

in which T represents degrees of temperature on the absolute scale (§ 16) and Z is a factor or multiplier constant or variable, but determinate, with which the temperature is to be multiplied in order to produce the desired or observed value for the first member, with any selected medium. V is the volume at the end of the stroke, through which the piston has swept, and P the pressure at the end of the stroke and which has prevailed throughout it if the pressure was constant, or is the mean pressure if the latter was variable. All solids are at once thrown out as media, because they lack the property of any considerable range of volume except in the form of coiled springs which are not available where heat energy is the motor energy, and consideration can be confined to media in liquid and gaseous states. Liquids in the liquid state are thrown out for the same reasons: the change of volume by heat alone, *if they remain liquids*, is too small to

make them useful. Gases, on the other hand, have the property of undergoing wide ranges of volume, and of experiencing considerable changes of condition for small changes of temperature. Their great elasticity enables the gases to be conveniently stored in considerable weights in bulks which are not inconvenient; and when liberated from the motor-cylinder after their work is done they pass out easily, and cause the least negative pressure or effort to expel them. It will furthermore be evident that, other things being equal, a medium which has a high value for the factor Z will make a more powerful motor with a given size of cylinder than one which has a low value for that factor. It goes without saying that the greater the value for the quantity of heat in units which is brought into the cylinder and utilized there per stroke, the more powerful and economical is the motor for its size or bulk. Shall now the gases to be selected as media be permanent or condensable gases?

It must not be overlooked that the volume of gas which has filled the volume V of the cylinder at the end of the completed stroke must be expelled therefrom on the return of the piston to its starting-point. The effort necessary to do this work of expulsion is a charge upon the net or effective work outside of the cylinder, because it must be subtracted from the gross or driving effect of the working-pressure medium, and it is of advantage to make it as small as possible in the interests of size of engine for a given effective power, and for the sake of reducing ineffective effort on general principles. Now with the permanent gases, the best which can be done is to open the cylinder-volume by generous passages and valves to a larger and cooler volume in which by the lowering of T and the increase of V, the value of P shall be lowered as far as it can be done for that gas. For now a definition can be made of a permanent gas which shall be more exact and definite than that of the preceding paragraph. A permanent gas is one in which the value for the multiplier Z

determined experimentally for one condition of pressure volume and temperature is true for all other natural conditions, or in which Z is a constant. The equation is then usually written

$$PV = RT,$$

and values of R for different media will be given shortly. With the condensable or non-permanent gases a relation of pressure and temperature exists in which the gas changes to a liquid with a very great diminution of volume at that period of change, and of course a very great drop in the value for P. If then a condensable gas be used, and it be convenient at the end of the working or forward stroke to establish the conditions under which the gas goes back to liquid, the negative pressure for the expulsion stroke drops much further than it can conveniently be made to do with the permanent gases. If then the engine works without condensation of its medium, it makes little difference whether the gas be permanent or not, but the value for Z is the important primary matter; if the engine can work with condensation, the use of the condensable gases gives a smaller engine. The condensable medium, however, must be so chosen that the appliances for its condensation and revaporization shall not be inconvenient to an extent which may offset its advantage.

It is, however, not enough to have the value of the factor Z or R large in the above formula. It is obvious that if the equation is a true one it will hold for all values of V, and will be true for a volume of one cubic foot. If D denote the weight per cubic foot or the density of the medium, then these must vary inversely as each other, or $v = \dfrac{1}{D}$. Then the formula will be written

$$\frac{P}{D} = RT,$$

an expression in which the volume does not appear, and which states that the density of the gas must diminish as the temperature is made to increase, and when R is large the density must diminish faster than the pressure rises for such a medium. Hence the conclusion that a desirable medium is one in which the changes of temperature within a given range do not produce wide differences in specific gravity. Such media must cause the change in T to cause changes in pressure, which is the thing sought for.

The equation

$$\frac{P}{D} = RT$$

permits of an interesting extension of its discussion. If the equation as written is true for air, it will take for any other permanent gas whose density is D' and which has for its factor a quantity represented by R' a form

$$\frac{P}{D'} = R'T.$$

Dividing these equalities member by member,

$$\frac{D'}{D} = \frac{R}{R''}$$

or the factors R and R' will vary inversely as the weights per cubic foot or the densities. These latter are usually well known and of easy access in tables, from which the values of R can be found. If the densities are given with air as a standard, then

$$\frac{D'}{D} = \text{specific gravity} = S.$$

From this R' can be found by dividing R for air by S; or

$$R' = \frac{R}{S}.$$

The following table gives such determinations made by this method:

	Spec. Grav. *S.*	*R* Centigrade.	*R* Fahrenheit.
Atmospheric air....................	1.00000	96.0376	53.354
Nitrogen..........................	.97137	98.867	54.926
Oxygen...........................	1.10563	86.862	48.257
Hydrogen.................	.06926	1386.579	770.322
Carbonic acid	1.52901	62.808	34.895
Steam-gas (ideal, Rankine).........	.62209	154.379	85.766
" " (" Zeuner)..........	.62300	154.153	85.641
Steam, saturated (Zeuner).........	.64000	150.160	83.422

Much effort has been directed towards securing a medium which should pass from a liquid to a gaseous state with least absorption of heat in such vaporization process, so as to secure a high vapor tension or pressure of the gas in its generator with a low specific heat or heat in the liquid when in a state to make itself into a gas. The difficulty so far encountered in all these attempts has been an inherent one: that the vapors from the volatile liquids which heat easily are so much more dense than the vapor of water with which they compete and are compared, that just about as much greater weight of the substituted liquid has to be vaporized as appears to be saved by the lower temperature of vaporization if the same mechanical energy is developed at the piston.

Another way of stating the same truth or result is, that to carry into the working cylinder as much heat as possible per stroke is one of the objects sought in a medium, and to have it carry out with it the least possible heat is the other. The volatile vapors with low specific heat compel a large weight of their substance to be used to carry into the cylinder a great number of heat-units, and it is not easy to prevent their carrying too many out of the engine with them, since, unless great quantities of cool condensing water are used, these easily vaporizable media refuse to return back to liquids. If they do not return to liquids, they carry away

the heat used to vaporize them, which is lost; and the work of handling great weights of water for condensing more than offsets the apparent gain.

108. Some Heat-carriers which have been used as Media in Heat-engines.—The two media most used in heat-engines are steam, which is the gaseous state of water, and air. These are accessible, cheap, safe, innoxious, odorless, non-inflammable.

Other media may be mentioned:

> Ammonia (NH_3).
> Acetone (C_3H_6O).
> Alcohol (C_2H_6O).
> Bisulphide of carbon (CS_2).
> Chloride of carbon (CCl_4).
> Chloroform ($CHCl_3$).
> Ether ($C_4H_{10}O$).
> Naphtha and Gasoline (C_6H_{14} to C_8H_{18}).

These are all more volatile than water, or make a vapor at a lower temperature; but they are costly to buy and hence must be condensed after working in the cylinder, and require for this an excess of cool condensing water. Many of them have an odor, some an offensive one; some are inflammable, some explosive, some irrespirable.

The objection to air and the other permanent gases is the high range of temperature under which it must work, and the high initial pressures corresponding to such high temperatures, and yet withal the low mean pressure which must follow expansive working of the air. There can be no condensation and hence the working cylinder has to be bulky. This will be illustrated in detail later. Some of the properties of heat media are exhibited in the table on page 173, and others will be found in appendices under their respective heads. The only objection to steam as a heat-carrier is its possession of a property whereby the withdrawal of heat

while doing work during expansion in the cylinder causes a
condensation of some of the steam, resulting in the formation
of a mist which has an enormous absorptive capacity for heat
and causes the phenomenon of cylinder condensation, to
which later reference will be made.   This very property,
however, gives steam an advantage as far as size of cylinder
is concerned.

109. **Vapors.**—The term vapor which has been used
above is often employed to define the gaseous state of a body
which is commonly a liquid within ordinary ranges of pressure
and heat.   In this sense steam is the vapor of water, and any
condensable gas should be called a vapor.   This is the
proper use of the term.   It is often, however, loosely applied
to define a gaseous body having or carrying finely subdivided
liquid particles in it which do not combine into drops, but
give an opacity or visibility to the mixture of gas and liquid.
The white cloud of watery particles which appears to issue
from the exhaust-pipe of a steam-engine is not the true vapor
of water (steam is an invisible gas), but it is often called a
vapor when it should properly either be called vesicular
vapor or nebulous vapor or be known as a *mist*.   The term
vapor is often popularly used to cover those gases other than
steam which are used for motive-power purposes.

110. **Liquefaction, Fusion, or Melting. Latent Heat of
Fusion and Vaporization.**—If the notion of the mechanical
theory of heat be sound, it follows that a solid which has
become a liquid by the process of applying heat to it, as in
the melting of ice or sugar or tallow or lead or iron, must
have had its heat condition or heat energy greatly increased
by that process.   Very considerable quantities of heat in
units have been expended upon it, and yet the temperature
changes recognizable by the thermometer are not so very
great between the condition of hot solidity and that of incipi-
ent fluidity.   Conversely, a body like water, in passing to

THEORETICAL, ECONOMY OF VARIOUS VAPORS USED AS HEAT MEDIA IN A GIVEN ENGINE AT FIXED CUT-OFF, THE EXPANSION BEING SUFFICIENT TO CAUSE A FALL OF PRESSURE FROM 100 TO 14.7 LBS. PER SQUARE INCH. MEAN BACK-PRESSURE 14.7 LBS.

Substance used as a Heat Medium.	Specific Heat in Liquid Condition.	Temperature in Degrees Fahrenheit at pressures per square inch of		British Thermal Units necessary to be expended in order to generate 1 pound of vapor from "temp. of feed," equal to that given in column 4.	Efficiency or Fraction of Heat expended per col. 5, which can be realized as useful work in an engine.	Coal per hour per I. H. P., assuming the boiler to utilize 8000 heat-units per pound of coal—cylinder condensation neglected—in lbs.	Probable coal per hour per H. P., including allowance of ⅓d for cylinder condensation for saturated vapors, and zero for superheated vapors—in lbs.
		100 lbs.	14.7 lbs.				
1	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Steam..........	1.00	328	212	1002	0.14	2.30	3.10
Ammonia, saturated..	0.99	56	— 27	632	0.15	2.15	2.88
" superheated	0.99	282	8	940	0.13	2.48	2.48
Ether..........	0.55	215	95	195	0.15	2.15	2.88
Carbon bisulphide......	0.24	248	115	163	0.17	1.99	2.68
Air — worked as in Joule's engine (see Rankine's "Steam-engine," p. 373)......	0.24	607	70	160	0.46	0.70	0.70

the state of ice, requires the withdrawal of a large number of units of heat just at the freezing-point, to enable the change to occur. This is usually explained by saying that an increased energy is characteristic of the fluid state of that body over that attaching to it in its solid state; and that the heat or energy which disappears in effecting this change of state is used in overcoming molecular attractions. It is generally called *latent heat* (meaning concealed heat) because it is not recognizable as heat except when the reverse change occurs, although necessary to produce it. When the change is from a solid to a liquid, it is called the latent heat of fusion or liquefaction. The following table gives the accepted values determined by M. Person. The figures are the pounds of water which are raised one degree Fahrenheit by the release of heat when the bodies solidify; or the degrees Fahrenheit through which one pound of water would be raised by the same process. They must also be the same quantities for the lowering of the heat condition of water when its heat is demanded to liquefy the substance at the temperature of its fusion.

Water (ice)	140 to 142	Bismuth	22.726
Zinc	50.682	Sulphur	16.954
Silver	38.057	Lead	9.740
Tin	25.702	Phosphorus	9.018
Cadmium	24.588	Mercury	5.086

These figures, which are in B.T.U., must not be confounded with the temperatures of fusion already referred to in § 69.

When the change is from a liquid to a gas, the heat required for vaporization is called the latent heat of evaporation. This will be further discussed in a following chapter; but it will be apparent that media which have a high value for their latent heat of evaporation will carry more heat into the engine-cylinder than media in which this quantity is

smaller, and that upon usefully entrapping this large latent heat by condensation after the working stroke, the motor rejects less heat to waste than when the medium is reluctant to part with its latent heat or has none to give, as in the case respectively of the volatile vapors or the permanent gases used as media.

CHAPTER X.

PHYSICAL LAWS EXHIBITING THE EFFECTS OF HEAT UPON HEAT-CARRIERS.

111. Introductory.—The accepted principles of the mechanical theory of heat exact that when a body like a motor medium undergoes an increase of its heat energy, that increase shall be distributed to produce three effects:

1. An increase in the sensible temperature as discernible by thermometer or measurable by other means of observing actual energy.

2. An increase of volume, which means the doing of a certain amount of internal work upon the substance itself in overcoming the attractions of the particles for each other.

3. An overcoming of the forces exerted externally upon or against the body in its first state which have resisted the increase in volume, and which have therefore demanded an expenditure of energy in foot-pounds before the body could assume its greater bulk. If these three effects, each in foot-pounds, be represented respectively by the symbols A, B, and C, and Q denote the quantity of heat applied in heat-units, then it can be written that

$$A + B + C = 778Q,$$

or

$$Q = \frac{A + B + C}{778}.$$

In solids and liquids the quantity C will be so small as to be negligible, so that the expression can be written

$$Q = \frac{A + B}{778}.$$

In perfect gases there should be no force required to overcome the attraction of their particles and B will disappear; or, for such gases,

$$Q = \frac{A + C}{778}.$$

The above holds true, however, only within the limits of no change of state of the body or while it remains a solid, a liquid, or a gas. At the melting-point of a solid the quantity A disappears suddenly or gradually, and the addition of heat after that does not raise the temperature of the remainder of the solid or of that part which has become liquid until all has been melted. The additional heat is expended in increasing the quantity $B + C$. The same is true for liquids passing to vapors. These quantities require to be separately investigated for each material.

But for the permanent gases used for motive-power purposes, such as air, and for the most usual vapor, that of water, there are certain physical laws which are the result of experiment and analysis and which require to be studied. The permanent gases are the easiest to begin with and will be taken first.

112. Law of Gay-Lussac, or Charles' Law.—This may be stated: The increase of volume which a perfect gas receives when the temperature is increased 1° under a constant pressure of such gas is a fixed proportion of its initial volume at the temperature of *melting ice;* or, stated otherwise, *Equal increments of the volume of a perfect gas correspond very nearly to equal increments of its temperature as determined by*

a mercurial thermometer, provided the pressure is kept constant. The first statement has already been anticipated in the discussion of the air-thermometer and absolute temperature (§§ 15 and 16), and the values for the increment of volume for each thermometric degree as determined by Regnault and others; viz., $\frac{1}{273}$ or .00365 on the centigrade scale, and $\frac{1}{493}$ or .002035 on the Fahrenheit. Expressing this law in symbols, if $v_0 =$ an initial volume of any permanent gas at the temperature of melting ice, and if a represent the coefficient of increase for each degree of the thermometric scale, then the volume for any temperature t will be (at) times greater than the volume at melting-ice temperature on the centigrade scale, and $[a(t - t_0)]$ times greater on the Fahrenheit or other scale on which the reading at melting-ice temperature is not zero. This can be made general for all scales by calling t the *range of temperature* from melting ice as a starting-point, or

$$v = v_0(1 + at).$$

The coefficient a is practically or very nearly the same for all the permanent gases, air, oxygen, hydrogen, etc.

113. Coefficients of Expansion.—It has already been observed that the expansion of solid bodies by heat is so small a quantity within any normal range of temperature as to be of no moment in motive-power problems. The following tabular values from D. K. Clark will show the quantitative relations of certain materials. The figure is the length to be added to a unit length for each degree Fahrenheit.

Aluminium	.00001234	Lead	.00001571
Brass	.00001052	Plaster	.00000922
Bronze	.00000986	Silver	.00001079
Concrete	.00000795	Steel	.00000689
Copper	.00000887	Tin	.00001163
Iron, wrought	.00000648	Zinc	.00001407
" cast	.00000556		

Liquids expand between 32° and 212° F., with the volume at freezing as unity:

Water..................	1.0466	Nitric acid............	1.11
Salt solution...........	1.05	Oils...................	1.08
Mercury...............	1.0182	Ether and turpentine....	1.07
Alcohol...............	1.11	HCl and H_2SO_4.......	1.06

114. Law of Mariotte, or Boyle's Law.—The law of Mariotte, discovered by him in 1640, and announced by Robert Boyle in England independently at about the same date, may be stated: *The temperature of the gas remaining constant, the volumes of the same weight of gas at different pressures will be inversely as the pressures.*

Near the points of liquefaction of gases by pressure, departures occur from this law which are wider the nearer that point is reached, as the diminution of volume is then more than proportional to the increase of pressure,—as should be anticipated from the conditions. Expressing the law by symbols, if p_0 be an initial pressure expressed in any unit of pressure on a unit of area, and v_0 the corresponding initial volume of the gas, then for any other pressures and volumes which go together it will be true that

$$p_0 : p : : v : v_0;$$

or, more conveniently,

$$p_0 v_0 = pv = \text{a constant},$$

provided no change of temperature or heat energy occurs by reason of processes connected with such change of volume. It follows further, that since for a given weight of gas the density will vary inversely as the volume, the pressures must vary directly as the densities, and will be directly proportional to them at the same temperatures. Or, in symbols,

$$p_0 : p : : D_0 : D; \quad \text{or,} \quad \frac{p}{D} = \frac{p_0}{D_0} = \text{a constant}.$$

115. Combination of Mariotte and Gay-Lussac Law. Value of Symbol R.—It becomes simple and useful to combine the foregoing two laws so as to exhibit the behavior of a weight of gas undergoing change of volume and temperature under constant pressure, or change of pressure and temperature under constant volume. Let p_0, v_0, and t_0 be respectively the pressure volume and temperature of a given weight of gas at the temperature of melting ice. Then for a different pressure, p_1, let v_1 denote the corresponding volume, and t_1 the range of temperature attaching to the change of pressure to p_1, and let p_2, v_2, and t_2 denote the same quantities at a different pressure, p_2, and range of temperature t_2. It follows from Mariotte's law alone that

$$p_0 v_0 = p_1 v_1 = p_2 v_2.$$

But by the Gay-Lussac law the respective volumes for a range t_1 and t_2 belonging to the pressures p_1 and p_2 will be respectively

$$v_1 = v_0(1 + at_1);$$
$$v_2 = v_0(1 + at_2);$$

whence

$$p_1 v_1 = p_0 v_0(1 + at_1),$$

and

$$p_2 v_2 = p_0 v_0(1 + at_2).$$

Dividing one by the other, and transposing the factors p,

$$\frac{v_1}{v_2} = \frac{p_2(1 + at_1)}{p_1(1 + at_2)}.$$

Substituting for a its value in either thermometric scale ($\frac{1}{273}$ or $\frac{1}{493}$) and multiplying both numerator and denominator by it, we have

$$\frac{v_1}{v_2} = \frac{p_2}{p_1} \times \frac{273 + t_1}{273 + t_2},$$

or

$$\frac{v_1}{v_2} = \frac{p_2}{p_1} \times \frac{461 + t_1}{461 + t_2}.$$

But the last factor is the absolute temperature correspond-ing to the temperatures belonging to p_2 and p_1, respectively, so that the equations might be written, if a capital T denote the respective absolute temperatures,

$$\frac{v_1}{v_2} = \frac{p_2}{p_1} \times \frac{T_1}{T_2},$$

which may be again transformed so as to read

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}, \text{ and which must equal } \frac{p_0 v_0}{T_0},$$

which may be translated to say that *at constant pressure the volume will vary inversely as the absolute temperature, or at constant volume the pressure will vary inversely as the absolute temperature.*

It follows furthermore, that since for any permanent gas the quantity T_0, or the absolute temperature at the point of melting ice; the quantity v_0, or the volume occupied by a given weight of gas under atmospheric pressure at that tem-perature; and the quantity p_0, or the pressure on a unit of area corresponding to one atmosphere, are all known and evalu-ated, the product of them must be a constant for any known gas. It may be called R. Whence it will be true for any other conditions of pressure and volume which belong together that

$$PV = RT.$$

(Compare § 107.)

For air $R = 53.354$, since $p_0 = 2116.5$ pounds per square foot; $v_0 = \frac{1}{D_0} = \frac{1}{.080728}$, whose denominator is the weight of a cubic foot of dry air at the sea-level and 32° F., and

under one atmosphere of pressure called 14.7 pounds per square inch; and T_0 is 493 for Fahrenheit scale. For other gases, such as

Steam, superheated .... $R = 104.641$
Ammonia .... $R = 162.602$
Ether............................. $R = 107.830$
Alcohol........................... $R = 103.785$

It will be apparent from inspection of the term $\frac{p_0 v_0}{T_0} = R$ that it represents the outer work performed by a pound of gas when its temperature is raised one degree, or from 32° to 33° on the Fahrenheit scale. Let a cylinder be imagined of one square foot of area, in which fits a weightless piston loaded with a weight to represent one atmosphere or 14.7 × 144 = 2116.5 pounds, and enclosing below it one cubic foot of air. Let this cubic foot be expanded by heat to become two cubic feet. The work done will be 2116.5 × 1 = 2116.5 foot-pounds by one cubic foot, or

$$\frac{2116.5}{.080728} = 26217.66 \text{ foot-pounds}$$

by one pound of air. The denominator is the weight in pounds of one cubic foot. But to double the volume would require by the Mariotte law an expenditure of 493°; hence to expand through one degree would require but $\frac{1}{493}$ of that required to do the work of doubling the volume. Hence the outer work entailed by the rise of one degree temperature Fahrenheit will be

$$\frac{26217.66}{493} = \frac{p_0 v_0}{T_0} = 53.354 = R.$$

Similar calculation can be made for any gas, or R can be found by the other method discussed in § 107 by using the densities.

116. Specific Heat at Constant Pressure and at Constant Volume.—A most important consequence is suggested by the foregoing calculations and deductions concerning the symbol R in the equation $PV = RT$. If the volume of the gas is prevented from increase, the work represented by R is not disposed of against outside resistances, and will remain in the gas as heat not expended in work. By definition (§ 12) the specific heat of a substance is the amount of heat necessary to raise one pound of it through one degree Fahrenheit. This quantity must obviously be different for a gas which is free to expand and overcome the work represented by R, from the quantity which the gas takes when such work is not done upon outer resistances. Gases therefore have two specific heats: the specific heat at constant pressure, which may be represented by C_p, and the specific heat at constant volume, represented by C_v. The former is always the larger, since

$$C_p - C_v = \frac{R}{778}.$$

For air Regnault's experiments give for C_p 0.2375; for C_v 0.1691; whence

$$\frac{C_p}{C_v} = \frac{2375}{1691} = 1.408.$$

In any case where a gas is heated from a temperature absolute T_1 to another higher absolute temperature T_2 under a constant pressure, the work done will be that of overcoming the pressure through a space represented by the difference between the volume v_1 at the temperature T_1 and the volume v_2 which corresponds to the temperature T_2. Expressing this in symbols, the heat taken in will be, per pound of gas,

$$C_p(T_2 - T_1);$$

and the work done will be

$$p(v_2 - v_1), \text{ which must equal } R(T_2 - T_1).$$

The internal energy in the gas must be the net difference between these two quantities, or

$$(C_p - R)(T_2 - T_1).$$

When, on the other hand, the same weight of gas (one pound) was heated at constant volume from T_1 to T_2, it must be true that the heat taken in is expressed by

$$C_v(T_2 - T_1),$$

since no external work is done, and the whole applied heat energy goes to store up internal energy. But the same amount of heat energy was applied in the two cases; so that

$$C_v(T_2 - T_1) \text{ should equal } (C_p - R)(T_2 - T_1),$$
$$\text{or}$$
$$C_v = C_p - R,$$

as has just been shown above.

It may therefore be stated that the expression $C_v(T_2 - T_1)$ expresses or measures the change of internal energy in a unit weight of gas in changing its temperature from T_1 to T_2 in any manner, no matter how the volume or pressure may vary during the process. C_v has been called the real specific heat, and C_p the apparent specific heat.

117. Joule's Law.—A law determined experimentally by Joule, involving the foregoing determinations and extending them, may be stated as follows: *When a gas expands without doing work and without taking in or giving out heat (and therefore without changing its stock of internal energy), its temperature does not change.* This was proved by immersing two closed vessels in a vessel of water. They were connected by a tube with a cock in it. One was empty, and in the other was the gas at a considerable tension by compression. When the cock was opened, the gas expanded and equalized its pressure in the two vessels, but did no external work. The water surrounding the vessels underwent no change in temperature, but the cooling upon expansion was offset by the

warming effect in the other vessel. Hence it was inferred that the gas had neither gained nor lost heat, and, since it had done no work, the intrinsic energy was the same at the end as at the beginning, although both pressure and volume had undergone changes. Hence the conclusion that the intrinsic energy of a given weight of gas depends on its temperature only, and not on its pressure or volume; or, in other words, a change of pressure and volume not associated with a change of temperature leaves the internal energy unaltered. This has an important significance in operating with compressed air. Or, again, the same idea may be expressed by saying that the change of internal energy is independent of the relation of pressure to volume during a temperature change, but is dependent only upon the amount of such temperature change.

118. Graphical Representation of the Thermal Changes in a Gas.—Since the characteristic equation of a perfect gas ($pv = RT$) involves three factors which are variable and one constant factor, and of which variable factors one can be made an arbitrary to be assumed, it early attracted the attention of mathematicians that this equation was in the same form as that for a curve upon a surface whose points were given by their coordinates or perpendicular distances from three rectangular axes. If one factor or co-ordinate were assumed arbitrarily, the other two would give the relation between themselves on a plane surface, giving a curved plane figure; while if all three were variable the curve would be upon a surface whose section at every plane through it would be a curve.

This fact is of interest and significance in the field of speculative research, but by far the most usual cases are those in which one of the variables is assumed to be constant or to undergo no change, while the other two are varying according to the law of their relation for that particular gas. If the temperature, for instance, be assumed to be kept up by jack-

eting the working-cylinder with live steam or hot air, the changes designated by a properly drawn curve show the changes in pressure which take place as the volume is increased or diminished. This gives the form of diagram or curve which bounds the ideal indicator-diagram, taken or described by a pencil which records pressures as vertical lines, while the horizontal lines described from the engine cross-head are proportional to the volume created behind the piston by its motion from its dead-centre. By a similar process, the volume being kept constant for a given mass or weight of gas, a curve can be drawn showing the law of observed variation of pressure with temperature; or again, the pressure being kept constant, the law of variation of volume with temperature. This is a straight line, of course, whose equation is $v_1 = v_0 + at$ from the Gay-Lussac law (§ 112).

These various lines on one or another of the coordinate planes have received special names, some of which are as follows.

119. Lines of Constant or Equal Pressure. Isopiestic Lines, or Isobars.—When the change of condition in a gas is a change of its volume without change in its pressure, and
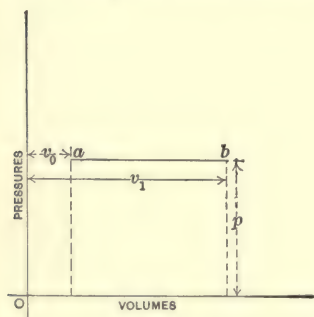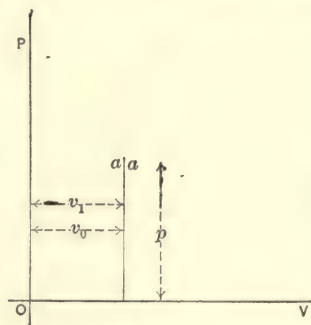

Fig. 43.   Fig. 44.

the same assumption is made as in the foregoing paragraph, that vertical ordinates represent pressures, and horizontal abscissas represent volumes, then a horizontal line ab (Fig. 43)

drawn at a height above the initial line of pressures at a distance proportional to that constant pressure and of a length proportional to the increase in volume (or decrease) will represent an isopiestic line. Such a line is drawn by the pencil of a steam-engine indicator when the piston moves in the cylinder and steam enters from the boiler by evaporation of the water within it without drop of pressure.

The temperature may or it may not change during the increase of volume. The diagram is silent on this point. The work done would be obviously

$$\text{Work} = p(v_1 - v_0).$$

120. Lines of Constant or Equal Volume. Isometric Lines.—When the pressure in the cylinder is increasing by addition of heat or pressure, while the volume occupied by the gas is not altered, a vertical line will represent such changes of pressure under the same suppositions as above. This is the line traced by the pencil of the indicator at the dead-centre of the piston-stroke when the valve has opened to admit steam behind the piston, but no motion has occurred to generate a volume in the cylinder to be filled. Since there is no volume swept through by the pressure, the work is zero (Fig. 44).

121. Lines of Constant or Equal Temperature. Isothermal Lines.—By the use of special appliances (steam-jacket, hot-air jacket, and the like) it is possible to supply to the weight or mass of gas enclosed in a working-cylinder the amount of heat which it is expending in the form of work both upon its own molecules in expanding, and the doing of the external work. In the case of a per-

Fig. 45.

manent gas acting in this way the pressure will fall as the
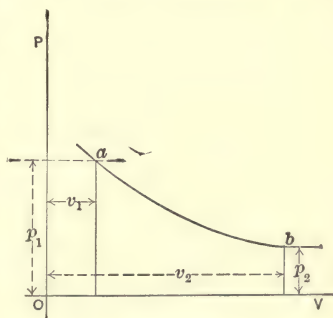
volume increases (Fig. 45), and in simple and inverse ratio to such increase because

$$\frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2} = R = \text{a constant,}$$

and the final temperature T_2 was the same as the initial T_1, by the hypothesis. In other words, the external work has received all the heat supplied to the working-gas from outside, and the intrinsic energy of the gas has remained constant, or undiminished by the doing of any work.

On the other hand, for a mixture of a liquid and its vapor, as in a steam-boiler, in which the temperature is kept constant by a continuous supply of heat from a fuel or a fire, the pressure of the combination of liquid and vapor remains constant; hence the isothermal for such a case will be a straight line, like the isobar or isopiestic line of Fig. 43. This is the condition in the admission line and the back-pressure lines of the indicator-diagram.

122. Isodynamic or Iso-energic Lines.—This name is applied to lines of a thermal diagram representing changes during which the intrinsic energy remains unaltered; that is, all the heat received is transformed into external work, and produces no change in the carrier during the process. The isothermal for a gas as above is also an isodynamic line, by definition.

123. Adiabatic Lines.—When the gas which is working by expansion within a cylinder overcomes the external resistance through a path, and is yet so contained within that cylinder that it can receive *no* heat from an outside source, it is apparent that in such expanding it should become cooled by the giving up of some of its intrinsic energy. The cylinder may be supposed to be absolutely non-conducting; hence no heat is transferred to or from the working-medium—which was supposed to happen in the isothermal working. Rankine gave the name *adiabatic* to this change of heat condition

(from *a*, not, and *diabainein*, to pass through); and adiabatic lines are those which represent the relation of volume to pressure during changes which occur without transmission of heat, as such, to the medium.

The adiabatic line is usually steeper than the isothermal which has a common point with it, as can be made apparent if the gas has a considerable change of volume with tem-

perature. In Fig. 46, if the curve *ii'* be an isothermal and start at *i* to represent the expansion which takes place *with* a transfer of heat to it during that process, it will be apparent that the curve *aa'* representing expansion *without* that added heat should have a less pressure when a final volume is reached which is the same for
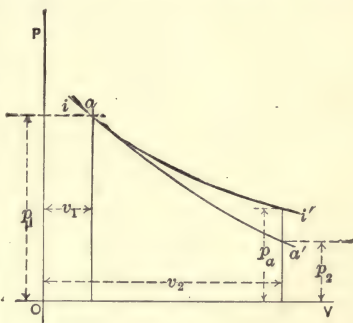
FIG. 46.

both. This will be equally true if the gas be compressed from a greater to a lesser volume. The isothermal curve must have heat withdrawn from the gas, thus diminishing its volume at the end of compression; adiabatic compression will leave in the gas the heat which corresponds to the work expended in such compression, and for the same final volume the pressure will be higher. It will be interesting in this connection to examine the table for the relative volumes of compressed air under adiabatic and isothermal compression given in Chapter XIII, § 182.

In the analytic representation of an adiabatic change it will no longer be true that $pv = RT$, but the equation for relation must be written

$$p_0 v_0{}^n = p v^n = \text{a constant};$$

in which the exponent *n* represents either a whole number or a fraction, but is constant for any one substance and is to be

experimentally determined.   The condition of true adiabatic
expansion (or compression) is rarely reached in practice,
because the cylinder-walls of metal castings such as are usual
must conduct heat in or out during the time in which the gas
or medium is changing its volume and pressure.   Such a
curve, however, with its exponent experimentally determined
is probably more nearly reached in the majority of cases than
the curve of the isothermal change in which n is unity.

124. **Isentropic Lines. Entropy.**—It would be aside
from the present purpose to review in detail the processes
used by the great mathematicians in their masterly deduc-
tions which resulted in the mathematical quantity which they
have called the '' thermodynamic function '' (Rankine) or
the '' entropy '' (Clausius).   In brief their reasoning may be
summarized by stating that the addition of heat to a body is
rendered evident by changes in pressure and volume.   These
simultaneous changes of infinitesimal extent give rise to an
equation of differential form which expresses the relation
between the ordinates and abscissæ for a given state, and
which it is desired to integrate by the methods of the calcu-
lus so as to express the law of change between certain finite
limits.   When the equation involves unknown functions of
the variable volume and pressure, and is in a general form,
the expedient has been used of introducing an integrating
factor.   If this factor is made itself a function of the pressure
and volume, the differential equation for a small increase in
heat H becomes

$$\delta H = y\delta x,$$

in which y is the reciprocal of the integrating factor and δx
is what the differential equation becomes when the equation
is thus made integrable.   Thence the investigation is con-
cerned with the labor of finding out which of the
functions of pressure and volume it is most useful to
assign to the factor y.   Subsequent research shows it to

be the absolute temperature, deduced from Carnot's prin-
ciple (§ 213), and therefore

$$\delta x = \frac{\delta H}{T}.$$

This is usually written by substituting ϕ for x,

$$\delta \phi = \frac{\delta H}{T},$$

whence

$$\delta H = T \delta \phi.$$

This is evidently a most elementary expression for an area,
and is to be used in that form hereafter in this treatise.

This factor ϕ was first segregated mathematically by
Clausius, and received from him its accepted name, *entropy*—
from the Greek words *en* and *tropē*, meaning a " turning
into " or a transformation. Its symbol in all languages is
the Greek letter ϕ. While it is of signal importance in heat-
engine discussions, it is impracticable to form a defensible
conception of the entropy as a property of heat media,
since it does not reveal itself to the senses nor to usual in-
struments of observation. A most helpful illustration or
analogue has been elaborated by Zeuner, Reeve, and others,
by the suggestion that the energy resident in a pound of water
to be used upon a water-motor is the product of the available
height or head above the motor, multiplied by the attraction
of gravitation upon the mass of the water. The head corre-
sponds to the temperature in heat-engines, and is the measure
of the availability of the medium when our lower temperature
level is fixed; the attraction of gravitation corresponds to
the entropy of the heat medium, which has been called its
" heat weight." We know as little about gravitation outside

of its phenomena and laws as we know about entropy, but this does not interfere with every-day applications and uses.

It is customary to take the value for the entropy factor as the difference between the value for the final stage and the initial stage of the expansion, each counted from freezing-point usually. This gives a finite value to be used as a factor, by methods to be discussed hereafter in more detail in Chapter XIV. For the present it will suffice to say that when heat is added to a permanent gas such as air,

$$\text{Entropy} = \int \frac{C\delta t}{T} = C \text{ hyp. log } \frac{T_1}{T_2},$$

which becomes equal to R hyp. log r when r is the ratio between the initial and final volumes or pressures or temperatures of the gas doing work by its expansion. This is made evident from the fact that the entire heat addition in such isothermal expansion appears as the external work; this external work being the expression RT hyp. log r (§ 166) can therefore be placed equal to $T\phi$.

In the case of steam-gas, working in a cylinder as dry saturated steam, the heat addition is that which has disappeared as latent heat; hence the entropy becomes

$$\frac{\text{Heat of vaporization of steam at } T}{T} = \phi = \text{entropy,}$$

exclusive of the heat addition made to the liquid from which the steam is formed. Hence for the condensable vapors the total entropy is made up of two parts. The difference between the entropy value of the liquid at the beginning and end of the heating process will be

$$\theta_1 - \theta_2 = C_v \int \frac{\delta t}{T_1} - C_v \int \frac{\delta t}{T_2} = C_v \text{ hyp. log } \frac{T_1}{T_2}$$

which indicates a progressive increase in the heat condition

from the lower temperature to the higher on the hypothesis that C_v, the specific heat of the liquid at constant volume, is the same at both ends of the range represented by the heating. (See columns 11 of § 136, and formulæ at its conclusion.) When the vapor begins to form, a part x has its state changed and receives the heat of vaporization r, or the latent heat. The sum of these is

$$\frac{x_1 r_1}{T_1} + \theta_1$$

for a state corresponding to T_1, and for a state T_2

$$\frac{x_2 r_2}{T_2} + \theta_2.$$

Hence the difference will be

$$\phi_2 - \phi_1 = \frac{x_2 r_2}{T_2} + \theta_2 - \theta_1 - \frac{x_1 r_1}{T_1},$$

or

$$\phi_2 - \phi_1 = \frac{x_2 r_2}{T_2} - \frac{x_1 r_1}{T_1} + C_v \text{ hyp. log } \frac{T_1}{T_2},$$

under the foregoing supposition.

If in the permanent gas the temperature has to be raised from a lower temperature T_0, the entropy above that at freezing will be made up of the heat to raise from T_0 to T_1, and in addition that to increase the pressure from that at p_0 to that at p_1. Hence the general expression becomes for a heating process not isothermal

$$\phi - \phi_0 = C_p \text{ hyp. log } \frac{T}{T_0} + (C_v - C_p) \text{ hyp. log } \frac{p}{p_0}.$$

When, on the other hand, the curve representing the relations of pressure and volume is no longer an isothermal,

but is an adiabatic, the condition must be met that by definition there is no longer a transfer of heat from without the medium. Hence in the equation

$$\frac{\delta H}{T} = \delta \phi$$

the quantity δH has become zero, so that while T remains finite the quantity $\delta \phi$ must become zero. This means that when the volume and pressure change according to the adiabatic law, the change in entropy for such changes is zero, or the entropy is constant for such changes of condition. Important deductions from this truth will be seen to follow hereafter (Chapter XIV), but for the present it may be observed that a curve representing an adiabatic variation of pressure and volume may be called a curve of constant entropy, or an isentropic line.

An interesting deduction can be made from the foregoing facts. If two isothermal lines represent two differing states of a heat medium at different times, the change which has made them differ is a change in temperature. Each curve represents by itself all variations of pressure and volume at a constant temperature, and the substance can therefore pass from one curve to the other curve only by having a temperature change brought in sufficient to make the transfer. If, on the other hand, two adiabatic lines represent two paths of variation of pressure with volume, with constant entropy attaching to each, but variable temperature, the heat-medium can only pass from being operated on one adiabatic to being operated on the other by such a heat-energy change as shall change the entropy of the medium. Temperature addition will only increase the relation of the pressure to the volume on either curve, but will not change the path of the process from one adiabatic to the other. Representing this graphically: if ii and $i'i'$ be two isothermals (Fig. 47) representing the

relations of pressure and volume of a gas in a cylinder each corresponding to its proper temperature T and T', it follows from what precedes that to have the product $p'v'$ belonging to T' become the product pv belonging to T, the property of the gas to be altered is its temperature. If, however, the curves aa and $a'a'$ in Fig. 48 are each an adiabatic, whose
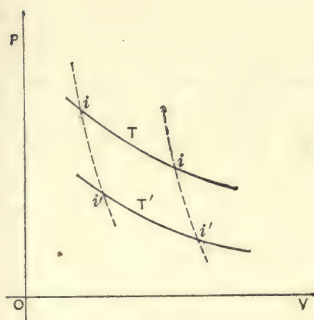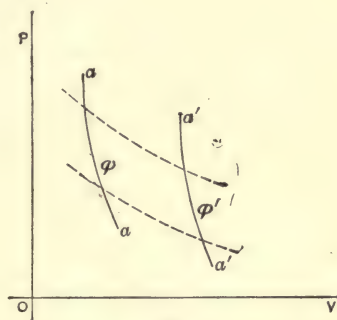
FIG. 47.   FIG. 48.

constant entropy is respectively ϕ and ϕ', the relation pv^n for one curve can only become the relation $p'v'^n$ proper for the other by a change in entropy which will correspond to the difference in that factor for the two states of the heat medium in question.

125. Plotting of Isothermal and Adiabatic Lines.—The graphical representation of the variation of pressure and volume isothermally is quite simple when the temperature is known or assumed and the value for R for the particular gas in question has been computed from known data. For, since

$$pv = RT = \frac{p_0 v_0}{T} T,$$

it appears that the curve of the isothermal is that of an equilateral hyperbola referred to the coordinate axes of zero

pressure and zero volume as asymptotes. The easiest way
is to calculate the product pv, and then to find the coordi-
nates of the vertex of the hyperbola, when $p = v$, by extract-
ing the square root of that product. When the vertex has
been found for the assumed value of T, other points are
found by making $2v = \frac{1}{2}p$, $4v = \frac{1}{4}p$, and so on (Fig. 49), or
the curve may be drawn by any reliable hyperbolograph.

A method much used to draw the hyperbola for the curve
of expansion on a pv diagram where the vertex does not come
at an observed point is required for indicator-diagrams. Any
point on the actual curve having been located, as B (Fig. 50),
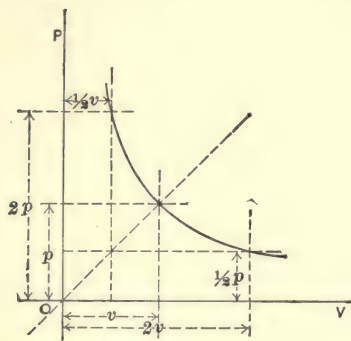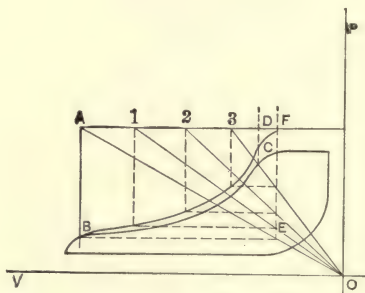
FIG. 49.   FIG. 50.

and the two coordinate axes of zero volume OP and zero pres-
sure OV having been established, a horizontal line is drawn at
a convenient pressure height above the greatest record of the
diagram. It may be at boiler-pressure, but this is not essen-
tial. From the point B draw horizontal and vertical con-
struction lines, and from the point A draw a diagonal to the
point O, the origin of coordinates; where the horizontal
through B intersects the diagonal, or at the point E, erect a
perpendicular EF, and complete the rectangle $ABEF$ by this
means. Then lines drawn from O and cutting EF and meet-
ing AF will determine other rectangles, and the intersections
of the lines p_1 and v_1, p_2 and v_2 will give in each case a point

on the curve. In indicator-diagram work the point B is usually taken at the point of release of steam at the end of the expansion, because at this point it is usually safe to assume that the steam is dry or vaporization is complete. The point C at which cut-off takes place or expansion begins can also be taken, although this is less certain and accurate, and necessarily locates a different hyperbola.

For the drawing of the adiabatic curve, the calculation must be made for the initial state with the relation of pressure and volume represented by $p_0 v_0{}^n$, in which the exponent n is either known or assumed. Then by the relations

$$p_0 v_0{}^n = p_1 v_1{}^n = p_2 v_2{}^n = \text{a constant}$$

points on the adiabatic curve are found. It will be brought out in a following chapter that for air n is usually 1.41, while for steam it is variously held to be 1.11 or 1.06

CHAPTER XI.

VAPORS AS HEAT-CARRIERS. STEAM.

130. Introductory.—It has been already said in Chapter IX that condensable vapors could also be used as heat-carriers, or those for which the expression $pv = RT$ was not true in all states. In the choice of a vapor medium there is room for a wider choice than among the permanent gases. The latter are so nearly alike in their physical qualities that the selection of the cheapest, most accessible, most innocuous, and most inexhaustible of the gases, which are the conditions which attach to air, have precluded any serious attempts to use any of the other gases. But with the vapors there is at once presented a wide variation in volatility, specific heat, vapor density, condensability, and behavior while doing the work of expansion which has attracted experiment and invention to attempt to utilize them. The vapor of water has the same convenient qualities on its practical side as are possessed by air among the gases. It is furthermore without disagreeable odor, is not inflammable nor explosive, and has the property of carrying great heat in small bulk, and of having the highest temperature at the convenient limits of pressure to be used in generators and working-cylinders. It must be shown therefore by any other vapor that it possesses advantages greater than the losses incidental to the use of steam if it is to seek to be used as a heat-carrier. In other words, the presumption is in favor of steam, and the burden of proof must lie with the rivals who may seek to displace it

(see Chapter XXI). There are, however, certain general facts and principles to be noted for all condensable vapors to which attention should be first directed before discussing the special properties of steam.

131. Saturated Vapor. Saturated Steam.—It should be a deduction from Chapter IX that in a vessel containing a given weight of a mixture of a vapor and the liquid from which the vapor is being given off by the application of heat at any given pressure, it will be impossible to change the value of the expression pv by changing the value of p without changing the proportion of liquid and vapor. If p is increased and the volume remain constant, some vapor will go back to liquid; if p is diminished, more vapor is formed. Furthermore, if heat is withdrawn from the mixture, condensation of vapor to liquid occurs; if heat is added, vaporization occurs. In other words, there is for every vaporizable liquid a temperature of vaporization (or a boiling-point) corresponding to every pressure. A vapor is said to be a saturated vapor when it is in that condition of pressure and temperature at which exists this equilibrium of tendency to vaporize and to condense. In the case of the steam in a boiler in which is the water being vaporized, the steam is saturated. Instantly, on the withdrawal of any steam into the cylinder, its place is supplied by fresh evaporation. When the throttle-valve is closed, either the generation of steam ceases, or if heat is still flowing into the water, the pressure at constant volume rises until generation is made to cease by a new equilibrium at such higher pressure. When the steam passes into a pipe at the temperature of saturation and leaves the boiler to do work in the engine, it is called *dry saturated* steam. Strictly, of course, it is the space which it occupies which is saturated, or which is filled with as much steam as it will hold *as steam* at that temperature. Any reduction of temperature from radiation or other heat-loss will cause some of the dry steam-gas to fall back to the state of water,

probably in a state of fine division or mist, or perhaps in the form of liquid hot water. Such steam is called *wet* steam; and wet is not synonymous with saturated. Heat applied to it to dry it by evaporating out the condensed water will cause it to become saturated once more.

When dry saturated steam enters its working-cylinder, and after-cut-off begins to work, increasing the volume it occupies and lowering its pressure by adiabatic expansion, the drop in temperature equivalent to the work done outside will cause a similar condensation of some steam unless the necessary heat is supplied from the cylinder-walls by jacket or otherwise to prevent a drop below the condition of saturation. The latent heat of vaporization must, however, be given out before this condensation occurs, and as this is a considerable proportion of the total heat (§ 138), the heat of the steam is strongly effective for doing work. By careful adjustment of the supply of heat to the jackets of a steam-engine, the curve of pressure and volume ratio can be made to be that of the saturation curve of steam, and the steam at the period of release from the cylinder will be dry saturated steam. It would otherwise be wet.

This matter will be further considered under paragraphs treating of cylinder-condensation and re-evaporation, and the negative specific heat of steam.

132. Superheated Vapor. Superheated Steam.—When steam or other vapor is raised by the application of additional heat to a temperature above that which belongs to its equilibrium of temperature and pressure when in contact with its liquid, it is called superheated. It is possible to superheat steam in contact with its liquid by reason of the slow transfer of heat through a large volume of gas, but as soon as convection or circulation occurs some of the liquid will be vaporized, and the state of saturation will be again established. Hence, as a rule, superheating is effected upon steam-gas which is isolated from its liquid, and in some other

chamber or vessel. The effect of superheating is to make it possible for some heat to be withdrawn from the superheated vapor before it passes to the saturation stage, after which further removal of heat results in condensation. This makes it a convenient expedient to diminish probable losses from the cylinder-condensation, hereafter to be discussed, since a range or margin above saturation-point must first be traversed before the inconveniences of active change to liquid set in.

Superheating occurs in effect when steam or other vapor passes through a narrow opening or a constriction of passage such as is caused by a partly-closed valve. On the one side is the higher potential of greater pressure, while upon the lower side is a volume too large to be filled as rapidly as it is created if the flow must be through the constricted passage. Hence a lower pressure prevails beyond the constriction, and yet no heat has been abstracted except that required to do the work represented by the friction of the vapor through the orifice. Hence the practical effect is to produce a steam in the space beyond the valve which is hotter than the temperature belonging to the pressure there—or which is, by definition, superheated; and it will behave as a superheated vapor in an engine-cylinder. The action or effect here spoken of is one of the advantages inhering in the throttling method of governing by means of a valve in the steam-pipe, and lies at the basis of a form of apparatus for measuring the amount of moisture present in a flowing current of wet steam in a pipe.

Superheated vapor may be made so hot that within the limits of its use in any machine it may behave as a permanent gas, and undergo no change of state. It may then be called steam-gas rather than a vapor. It loses some of its advantages as a heat-carrier when it is used as a permanent gas, and some practical difficulties result from its inconveniently high temperature.

133. Relations of Pressure and Temperature in Saturated Steam-vapor (Regnault).—The relation between pres-

sure volume and temperature for saturated steam and other vapors were investigated by the physicist Regnault at Paris with such thorough exhaustiveness that his results are the standards for all engineers. These researches are to be found in vol. XXVI of the " Mémoires de l'Institut de France," 1847. His method was to take the temperature of the boiling-point under varying pressures, the pressure being observed by a mercury-column, and the temperatures by mercury-thermometer. After corrections were applied and the readings reduced to the air-thermometer, the best results were plotted, a curve drawn through them, and an equation worked out whereby the relations desired could be calculated. His equation is of the form

$$\log p = a + b\alpha^n + c\beta^n,$$

in which p is the pressure in millimeters of mercury, and $n = t - t_0$, or is the range between the temperature at which the pressure p prevails and the lowest temperature at which the formula is true with the given values for the constants a, b, and c. Thus at a latitude of $45°$ the values for steam between $32°$ and $212°$ F. and in pounds per square inch

$$a = 3.025908;$$
$$\log b = 0.6117400;$$
$$\log c = 8.13204 - 10;$$
$$\log \alpha = 9.998181015 - 10;$$
$$\log \beta = 0.0038134;$$
$$n = t - 32;$$

while between $212°$ and $428°$ F. the quantities will be

$$a = 3.743976;$$
$$\log b = 0.4120021;$$
$$\log c = 7.74168 - 10;$$
$$\log \alpha = 9.998561831 - 10;$$
$$\log \beta = 0.0042454;$$
$$n = t - 212.$$

Fig. 51 presents to the eye a graphical representation on a small scale of the relations between temperature and pressure from the Regnault experiments, below 15 pounds pressure, and Fig. 97 hereafter will carry the relation up to 150 pounds.
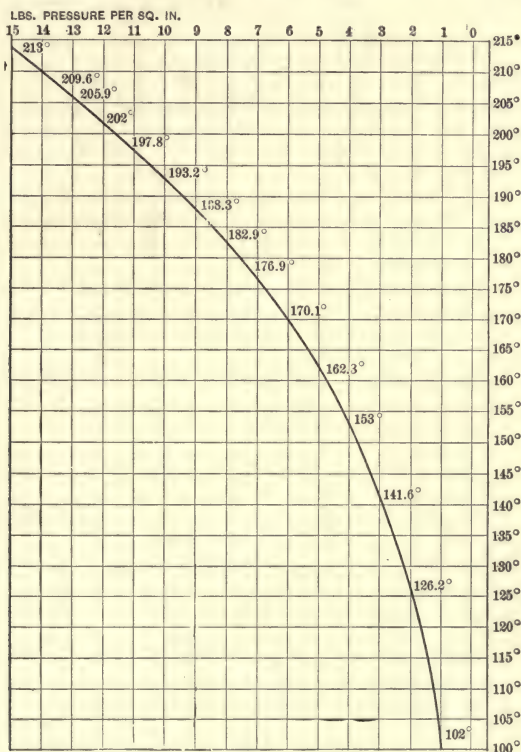
FIG. 51.

134. Rankine Formula for Pressure of Saturated Steam. —An approximate formula conforming very closely to Regnault's experiments was put forward by Rankine (*Edinburgh New Philosophical Journal*, July, 1849), and claimed by him to be sufficiently accurate between — 22° F. and 446° F. It is of the form

$$\log p = A - \frac{B}{T} - \frac{C}{T^2},$$

in which T is absolute temperature on the basis of $T = t° + 461°.2$, and for pounds per square inch

$$A = 6.1007;$$
$$\log B = 3.43642;$$
$$\log C = 5.59873.$$

For pounds per square foot $A = 8.2591$. The difference between the two authorities is given by the comparison in the following table:

Temperature Fahrenheit.	Pressure, in Pounds per Square Inch.	
	Regnault, at Latitude 45°.	Rankine Formula.
32	0.0890	0.083
77	0.4555	0.452
122	1.7789	1.78
167	5.579	5.58
212	14.697	14.70
257	33.711	33.71
302	69.27	69.21
347	129.79	129.80
392	225.56	225.9
428	336.26	336.3

135. Other Formulæ for Pressure and Temperature of Saturated Steam.—Many other formulæ have been worked out for the temperature and pressure of steam, having a certain accuracy and value within their several ranges.   Mallet's, ranging between 15 and 60 pounds pressure, is

$$p = \left(\frac{75 + t}{111.78}\right)^{6}$$

with centigrade degrees, and in pounds per square inch. Tredgold uses 175 as the denominator instead of 111.78.

Dulong and Arago give for pressures above 60 pounds per square inch, and centigrade degrees

$$p = (0.4873 + 0.012244t)^{5}.$$

Zeuner's formula is

$$pv = \frac{p_0 v_0}{T_0} T - C p^n,$$

in which by substitution is found

$$pv = 50.933 T - 192.50 p^{\frac{1}{4}},$$

in which p is in kilograms per square meter and v is the specific volume in kilograms per cubic meter.

De Volson Wood proposed

$$pv = 96.95 T - \frac{18500}{v^{0.22}},$$

in which p is in pounds per square foot, and v the cubic feet for one pound. For pounds per square inch, and v still in cubic feet per pound,

$$pv = 0.6732 T - \frac{128.5}{v^{0.22}}.$$

Other formulæ are by Dalton, Roche, Coriolis, and others.

136. Steam Tables.—Using the accepted formulæ and experiments of Regnault, Zeuner Rankine and others have computed tables giving the pressure corresponding to any temperature or the temperature corresponding to any pressure in common use. An abstract of such a table follows, giving also certain other data concerning steam which will be found useful and convenient. The pressure is given in even figures, as counted from a vacuum, and the corresponding temperature is therefore fractional. Following the computations by Prof. C. H. Peabody (the accepted authority in American practice), the value of the Fahrenheit freezing-point is taken as 492°.7. The reader and student is referred to Peabody's complete tables for further and fuller tabular information. The data for pressures above 300 pounds absolute are not reliable wherever the specific heat has entered as a

Pressure above Vacuum in Pounds per Sq. Inch.	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 pound in Cu. Feet.	Weight of 1 cu. ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B. T. U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
0.085	32	492.70	3336	.00030	1091.7	0	1091.7	...	52.4	0	2.215	−14.615
0.5	80	540.70	640.8	.00158	1106.3	48.04	1058.3	...	58.8	.095	2.052	−14.2
1	101.99	562.69	334.6	.00299	1113.1	70.0	1043.0	981.1	61.9	.1329	1.9845	−13.7
2	126.27	586.97	173.6	.00576	1120.5	94.4	1026.1	961.9	64.2	.1754	1.9235	−12.7
3	141.62	602.32	118.4	.00844	1125.1	109.8	1015.3	949.5	65.8	.2013	1.8869	−11.7
4	153.09	613.79	90.31	.01107	1128.6	121.4	1007.2	940.4	66.8	.2203	1.8612	−10.7
5	162.34	623.04	73.22	.01366	1131.5	130.7	1000.8	933.1	67.7	.2353	1.8416	−9.7
10	193.25	653.95	38.16	.02621	1140.9	161.9	979.0	908.4	70.6	.2842	1.7812	−4.7
11	197.78	658.48	34.88	.02866	1142.3	166.5	975.8	904.8	71.0	.2912	1.7730	−3.7
12	201.98	662.68	32.14	.03111	1143.6	170.7	972.9	901.5	71.4	.2976	1.7657	−2.7
13	205.89	666.59	29.82	.03355	1144.7	174.6	970.1	898.4	71.7	.3035	1.7588	−1.7
14	209.57	670.27	27.79	.03600	1145.8	178.3	967.5	895.5	72.0	.3091	1.7525	−0.7
14.7	212.00	672.70	26.42	.03794	1146.6	180.9	965.7	893.3	72.4	.3122	1.7477	0
15	213.03	673.73	26.15	.03826	1146.9	181.8	965.1	892.6	72.5	.3143	1.7467	+0.3
16	216.32	677.02	24.59	.04067	1147.9	185.1	962.8	890.0	72.8	.3192	1.7412	+1.3
17	219.44	680.14	23.22	.04307	1148.9	188.3	960.6	887.6	73.0	.3228	1.7351	2.3
18	222.40	683.10	22.00	.04547	1149.8	191.3	958.5	885.3	73.2	.3282	1.7313	3.3
19	225.24	685.94	20.90	.04786	1150.7	194.1	956.6	883.2	73.4	.3324	1.7269	4.3
20	227.95	688.65	19.91	.05023	1151.5	196.9	954.0	881.0	73.6	.3363	1.7224	5.3
25	240.04	700.74	16.13	.06199	1155.5	209.1	946.0	871.5	74.5	.3539	1.7039	10.3
30	250.27	710.97	13.59	.07360	1158.3	219.4	938.9	863.6	75.3	.3685	1.6890	15.3
35	259.19	719.89	11.75	.08508	1161.0	228.4	932.6	856.6	76.0	.3811	1.6765	20.3

Pressure in Pounds by Gauge.	Entropy of Vapor. ϕ_s	Entropy of Liquid. $\phi_w = \theta$	Heat Equiv. of External Work. $\frac{pu}{778}$	Heat Equiv. of Internal Work. r_1	Heat of Vaporization. r B.T.U.	Heat of Liquid. q	Total Heat. λ	Weight of 1 cu. ft. in Pounds.	Volume of 1 pound in Cu. Feet.	Temperature, Degrees Absolute.	Temperature, Degrees Fahr.	Pressure above Vacuum in Pounds per Sq. Inch.
1	12	11	10	9	8	7	6	5	4	3	2	1
24.3	1.6657	.3921	76.7	850.3	927.0	236.4	1163.4	.09644	10.37	727.83	267.13	40
30.3	1.6564	.4020	77.2	844.8	922.0	243.6	1165.6	.1077	9.297	734.99	274.29	45
35.3	1.6480	.4109	77.7	839.7	917.4	250.2	1167.6	.1188	8.414	741.55	280.85	50
40.3	1.6404	.4191	78.2	834.9	913.1	256.3	1169.4	.1299	7.696	747.59	286.89	55
45.3	1.6339	.4267	78.6	830.7	909.3	261.9	1171.2	.1409	7.096	753.21	292.51	60
50.3	1.6275	.4337	79.0	826.5	905.5	267.2	1172.7	.1519	6.583	758.47	297.77	65
55.3	1.6218	.4402	79.4	822.7	902.1	272.2	1174.3	.1628	6.144	763.41	302.71	70
60.3	1.6165	.4464	79.7	819.1	898.8	276.9	1175.7	.1736	5.762	768.08	307.38	75
61.3	1.6156	.4476	79.8	818.4	898.2	277.8	1176.0	.1757	5.691	768.98	308.28	76
62.3	1.6144	.4487	79.9	817.6	897.5	278.7	1176.2	.1779	5.621	769.88	309.18	77
63.3	1.6135	.4499	79.9	817.0	896.9	279.6	1176.5	.1801	5.554	770.76	310.06	78
64.3	1.6126	.4511	80.0	816.3	896.3	280.5	1176.8	.1822	5.488	771.64	310.94	79
65.3	1.6116	.4522	80.1	815.5	895.6	281.4	1177.0	.1843	5.425	772.50	311.80	80
66.3	1.6106	.4534	80.1	814.9	895.0	282.3	1177.3	.1865	5.362	773.36	312.66	81
67.3	1.6097	.4545	80.2	814.2	894.4	283.2	1177.6	.1886	5.301	774.21	313.51	82
68.3	1.6087	.4557	80.3	813.4	893.7	284.1	1177.8	.1908	5.240	775.06	314.36	83
69.3	1.6078	.4568	80.3	812.8	893.1	285.0	1178.1	.1930	5.182	775.89	315.19	84
70.3	1.6069	.4579	80.4	812.1	892.5	285.8	1178.3	.1951	5.125	776.72	316.02	85
71.3	1.6067	.4590	80.4	811.5	891.9	286.7	1178.6	.1973	5.069	777.54	316.84	86
72.3	1.6052	.4601	80.5	810.8	891.3	287.5	1178.8	.1994	5.014	778.35	317.65	87
73.3	1.6043	.4612	80.5	810.2	890.7	288.4	1179.1	.2016	4.961	779.15	318.45	88
74.3	1.6033	.4622	80.6	809.5	890.1	289.2	1179.3	.2037	4.909	779.95	319.25	89

Pressure in Pounds by Gauge.	Entropy of Vapor. ϕ_s	Entropy of Liquid. $\phi_w = \theta$	Heat Equiv. of External Work. $\frac{Aw}{778}$	Heat Equiv. of Internal Work. r_1	Heat of Vaporization. r B. T. U.	Heat of Liquid. q	Total Heat. λ	Weight of 1 cu. ft. in Pounds.	Volume of 1 pound in Cu. Feet.	Tempera-ture, Degrees Absolute.	Tempera-ture, Degrees Fahr.	Pressure above Vacuum in Pounds per Sq. Inch.
1	12	11	10	9	8	7	6	5	4	3	2	1
75.3	1.6027	.4633	80.7	808.9	889.6	290.0	1179.6	.2058	4.858	780.74	320.04	90
76.3	1.6018	.4643	80.7	808.3	889.0	290.8	1179.8	.2080	4.808	781.53	320.83	91
77.3	1.6009	.4653	80.8	807.6	888.4	291.6	1180.0	.2101	4.760	782.30	321.60	92
78.3	1.6001	.4663	80.8	807.1	887.9	292.4	1180.3	.2122	4.712	783.07	322.37	93
79.3	1.5992	.4673	80.9	806.4	887.3	293.2	1180.5	.2144	4.665	783.84	323.14	94
80.3	1.5984	.4683	80.9	805.8	886.7	294.0	1180.7	.2165	4.619	784.59	323.89	95
85.0	1.5947	.4733	81.2	802.8	884.0	297.9	1181.9	.2271	4.403	788.28	327.58	100
90.0	1.5909	.4780	81.4	799.9	881.3	301.6	1182.9	.2378	4.206	791.83	331.13	105
100.0	1.5842	.4869	81.9	794.4	876.3	308.7	1185.0	.2589	3.862	798.56	337.86	115
105.0	1.5812	.4911	82.1	791.9	874.0	312.0	1186.0	.2695	3.711	801.75	341.05	120
120.9	1.5724	.5027	82.6	784.7	867.3	321.4	1188.7	.3009	3.323	810.73	350.03	135
125.0	1.5697	.5064	82.8	782.3	865.1	324.4	1189.5	.3113	3.212	813.55	352.85	140
135.0	1.5648	.5133	83.1	778.1	861.2	320.4	1191.2	.3221	3.011	818.96	358.26	150
150.0	1.5581	.5230	83.6	772.0	855.6	338.0	1193.6	.3635	2.751	826.58	365.88	165
160.0	1.5538	.5290	83.8	768.2	852.0	343.0	1195.0	.3841	2.603	831.35	370.65	175
165.0	1.5518	.5319	83.9	766.4	850.3	345.4	1195.7	.3945	2.535	833.67	372.97	180
170.0	1.5498	.5347	84.0	764.6	848.6	348.2	1196.4	.4049	2.470	835.93	375.23	185
175.0	1.5492	.5375	84.1	762.9	847.0	350.1	1197.1	.4153	2.408	838.14	377.44	190
180.0	1.5401	.5402	84.2	761.1	845.3	352.4	1197.7	.4257	2.349	840.31	379.61	195
185.0	1.5435	.5429	84.3	759.5	843.8	354.6	1198.4	.4359	2.294	842.43	381.73	200
190.0	1.5426	.5454	84.4	757.8	842.2	356.8	1199.0	.4461	2.241	844.52	383.82	205
195.0	1.5410	.5480	84.5	756.2	840.7	358.9	1199.6	.4565	2.190	846.57	385.87	210

Pressure above Vacuum in Pounds per Sq. Inch	Temperature, Degrees Fahr.	Temperature, Degrees Absolute.	Volume of 1 pound in Cu. Feet.	Weight of 1 cu. ft. in Pounds.	Total Heat. λ	Heat of Liquid. q	Heat of Vaporization. r B. T. U.	Heat Equiv. of Internal Work. r_1	Heat Equiv. of External Work. $\frac{pu}{778}$	Entropy of Liquid. $\phi_w = \theta$	Entropy of Vapor. ϕ_s	Pressure in Pounds by Gauge.
1	2	3	4	5	6	7	8	9	10	11	12	1
215	387.88	848.58	2.142	.4669	1200.2	361.0	839.2	754.6	84.6	.5504	1.5393	200
225	391.79	852.49	2.051	.4876	1201.4	365.1	836.3	751.6	84.7	.5553	1.5363	210
240	397.41	858.11	1.928	.5186	1203.2	371.0	832.2	747.3	84.9	.5621	1.5319	225
250	400.99	861.69	1.854	.5393	1204.2	374.7	829.5	744.5	85.0	.5665	1.5292	235
260	404.47	865.17	1.785	.5601	1205.3	378.4	826.9	741.7	85.2	.5707	1.5264	245
270	407.85	868.55	1.722	.5809	1206.3	381.9	824.4	739.2	85.2	.5748	1.5239	255
280	411.12	871.82	1.662	.6020	1207.3	385.3	822.0	736.7	85.3	.5787	1.5215	265
290	414.32	875.02	1.607	.6220	1208.3	388.6	819.7	734.3	85.4	.5826	1.5193	275
300	417.42	878.12	1.554	.6440	1209.3	391.9	817.4	732.0	85.4	.5863	1.5171	285
350*	432.0	892.70	1.325	.7545	1213.7	406.3	807.5	721.7	85.8	.5878†	1.4923	335*
400	444.9	905.60	1.167	.8572	1217.7	419.8	797.9	711.5	86.4	.6043	1.4853	385
450	456.6	917.30	1.042	.9595	1221.3	432.2	789.1	702.4	86.7	.6152	1.4754	435
500	467.4	928.10	0.942	1.0620	1224.5	443.5	781.0	693.9	87.1	.6313	1.4728	485
550	477.5	938.20	0.859	1.1640	1227.6	454.1	773.5	686.2	87.3	.6419	1.4663	535
600	486.9	947.60	0.790	1.2660	1230.5	464.2	766.3	678.6	87.7	.6523	1.4609	585
700	504.1	964.80	0.680	1.4700	1235.7	482.4	753.3	665.2	88.1	.6678	1.4485	685
800	519.6	980.30	0.597	1.6740	1240.3	498.9	741.4	653.2	88.2	.6831	1.4393	785
900	533.7	994.40	0.532	1.8780	1244.7	514.0	730.6	642.3	88.3	.6981	1.4328	885
1000	546.8	1007.50	0.480	2.0820	1248.7	528.3	720.3	631.5	88.8	.7129	1.4278	985

* From Buel's Tables above 350 pounds.   † Not reliable.   Specific not known at these temperatures, and assumed to be unity.

factor, since its value at these higher pressures has not been fixed.

The columns after No. 5 have been calculated as follows:

$$\lambda = 1091.7 + 0.305(t - 32);$$

$$q = 1 + 0.00004t + 0.0000009t_2, \text{ in centigrade units,}$$
reduced to Fahr.;

$$r = \lambda - q = \text{Col. 6} - \text{Col. 7};$$

$$r_1 = r - \frac{pu}{778} = \text{Col. 8} - \text{Col. 10};$$

Col. 10 = (Col. 1 $\times$ 144) $\times$ Col. 4 $\div$ 778;

Col. 11 = Spec. Ht. $\times$ Hyp Log $\dfrac{T}{T_0}$;

Col. 12 = $\dfrac{\text{Column 8}}{\text{Column 3}}$ + Column 11; or $\phi_s = \dfrac{L}{T} + \phi_w$.

137. Saturated Vapor Pressures and Temperatures for Media other than Steam.—In the Comptes Rendus of the Academie des Sciences, Tome xxxvi, will be found the formulæ elaborated by Regnault to express the relations between temperature centigrade and pressure by mercury-column for other vapors such as alcohol, ether, chloroform, carbon bisulphide, and carbon tetrachloride, all of which have attracted experimenters who have wished to replace the vapor of water by another medium (see also Chapter XXI). His equations and constants are given in the following table:

PRESSURE OF SATURATED VAPORS.

Quantity.	Alcohol.	Ether.	Chloroform.	CS_2	CCl_4.
Log p	$a - ba^n + c\beta^n$	$a + ba^n - c\beta^n$	$a - ba^n - c\beta^n$	$a - ba^n - c\beta^n$	$a - ba^n - c\beta^n$
a	5.4562028	5.0286298	5.2253893	5.4011662	12.0962331
Log a......	$\bar{1}.99708557$	0.0145775	$\bar{1}.9974144$	$\bar{1}.9977628$	1.9997120
b......	4.9809960	0.0002284	2.9531281	3.4405663	9.1375180
Log β......	$\bar{1}.9409485$	$\bar{1}.996877$	$\bar{1}.9868176$	$\bar{1}.9911997$	$\bar{1}.9949780$
c	0.0485397	3.1906390	0.0668673	0.2857386	1.9674890
η .:	$t + 20$	$t + 20$	$t - 20$	$t + 20$	$t + 20$
Limits (C.).	$- 20, + 150°$	$- 20°, + 120°$	$+ 20°, + 164°$	$- 20°, + 140°$	$- 20°, + 188°$

138. Total Heat of Steam.—In the case of a hot liquid and its vapor, there will be a certain amount of heat resident in the liquid which has been required to raise it to the point at which the vapor forms at that pressure; there will be, furthermore, the heat represented by the internal work of disgregation of the particles of liquid to change its state to vapor (§ 111) and the heat which has disappeared in overcoming the external pressure. The usual expression for this sum (which is called the total heat of the vapor) is counted from the freezing-point of water, and for a unit of weight is the heat required to raise that weight of water from freezing-point to a given temperature, and evaporate it completely into steam at that temperature. Science is indebted also to Regnault's investigations for the accepted expression for this total heat, which has the form

$$\lambda = A + Bt,$$

in which the constants in centigrade units give

$$\lambda = 606.5 + 0.305t;$$

and in Fahrenheit scales

$$\lambda = 1091.7 + 0.305(t - 32°).$$

For the same other vapors as in the preceding paragraph, Regnault's figures for the total heat are, for

Ether .... $\lambda = 94 + 0.45t - 0.00055556t^2$
Chloroform...... $\lambda = 67 + 0.1375t$
CS_2............. $\lambda = 90 + 0.14601t - 0.0004123t^2$
CCl_4........... $\lambda = 52 + 0.14625t - 0.000172t^2$
Aceton.......... $\lambda = 140.5 + 0.36644t - 0.000516t^2$

139. Heat of the Liquid.—If the specific heat of water be considered unity at all ranges of temperature, the heat of the

liquid water would be the same as its range above freezing-point for each pound. At lower temperatures, the specific heat is unity. Regnault found that from 0° to 100° C. it was 1.005 and from 0° to 200° it was 1.016. Assuming an equation of the form

$$q = 1 + At^2 + Bt^3,$$

and finding the values for the constants by aid of known observed values of q, the usually accepted equation results:

$$q = t + 0.00002t^2 + 0.0000003t^3.$$

This can be used to work back to the specific heat by the methods of the calculus, since the differential of the value of q taken with respect to the temperature will be the heat required to produce this differential change, which is the specific heat; or

$$C = \frac{dq}{dt} = 1 + 0.00004t + 0.0000009t^2.$$

As before, the heat of the liquid for the other vapors investigated by Regnault gives values for q as follows:

<center>HEAT OF THE LIQUID q.</center>

Alcohol.............. $0.54754t + 0.0011218t^2 + 0.000002206t^3$
Ether............... $0.52901t + 0.0002959t^2$
Chloroform.......... $0.23235t + 0.0000507t^2$
CS_2................. $0.23523t + 0.0000815t^2$
CCl_4............... $0.19798t + 0.0000906t^2$
Aceton, C_3H_6O....... $0.50643t + 0.0003965t^2$

140. Heat of Vaporization. Internal Latent Heat.— It is made apparent from the foregoing that if the heat of the liquid be subtracted from the total heat, the remainder will be the heat absorbed in the vaporization process. If this be designated by the symbol r

$$r = \lambda - q.$$

It may be seen, however, that r is really made up of the sum of two quantities. One is the internal latent heat, or heat of disgregation, and the other is the heat equivalent to the work done by a unit liquid volume expanding into vapor volume u against a pressure p. If then r_1 denote the excess of the heat in units of heat which is contained in the vapor over the heat contained in the unit of weight of the water at 32° F. from which the vapor was produced, the above equation can be written

$$\lambda = q + r_1 + \frac{pu}{778},$$

since the last term will express the heat which has been expended in increasing the volume from a smaller bulk s to a larger bulk σ, which we may call U, and overcoming the pressure P which must be constant during the evaporation.

An empirical formula in the French system for r_1, proposed by Zeuner, gives the following values, which are fairly approximate:

<div align="center">INTERNAL LATENT HEAT r_1.</div>

Water	$575.40 - 0.791t$
Ether	$86.54 - 0.10648t - 0.0007160t^2$
Chloroform	$62.44 - 0.11282t - 0.0000140t^2$
CS_2	$82.79 - 0.11446t - 0.0004020t^2$
CCl_4	$48.57 - 0.06844t - 0.0002080t^2$
Aceton	$131.63 - 0.20184t - 0.0006280t^2$

141. Specific Volume of Hot Liquids.—For the accepted data concerning the increase of volume of the liquid which expands by heat (although much less than the expansion in change of state) the best values are those given by Hirn in the Annales de Chimie et de Physique for 1867, as the result of experiments using the liquids in question as the registering medium as in a thermometer. Usually the expansion of the liquid in connection with its vapor can be neglected in comparison with the expansion of the vapor, without detectable

HEAT AND HEAT-ENGINES.

error. Or, in other words, the value for σ is considered to be constant. Hirn's data are as follows:

$\sigma =$ SPECIFIC VOLUMES OF HOT LIQUIDS.

		Logarithms.
Water, 100° C. to 200° C. (Vol. at 4° = unity.)	$v = 1 + 0.00010867875t$ $+ 0.0000030073653t^2$ $+ 0.000000028730422t^3$ $- 0.0000000000066457031t^4$	$6.0361445 - 10$ $4.4781862 - 10$ $1.4583419 - 10$ $8.8225409 - 20$
Alcohol, 30° C. to 160° C. (Vol. at 0° = unity.)	$v = 1 + 0.00073892265t$ $+ 0.00001055235t^2$ $- 0.0000000092480842t^3$ $+ 0.000000000040413567t^4$	$6.8685991 - 10$ $3.0233492 - 10$ $2.9660517 - 10$ $0.6065278 - 10$
Ether, 30° C. to 130° C. (Vol. at 0° = unity.)	$v = 1 + 0.0013489059t$ $+ 0.0000065537t^2$ $- 0.000000034490756t^3$ $+ 0.00000000033772062t^4$	$7.1299817 - 10$ $4.8164866 - 10$ $2.5377028 - 10$ $0.5285571 - 10$
Carbon Bisulphide, 30° C. to 160° C. (Vol. at 0° = unity.)	$v = 1 + 0.0011680559t$ $+ 0.0000016489598t^2$ $- 0.0000000081119062t^3$ $+ 0.0000000000060946589t^4$	$7.0674636 - 10$ $4.2172103 - 10$ $0.9091229 - 10$ $9.7849494 - 20$
Carbon Tetrachloride, 30° C. to 160° C. (Vol. at 0° = unity.)	$v = 1 + 0.0010671883t$ $+ 0.0000035651378t^2$ $- 0.0000000149492581t^3$ $+ 0.0000000000851823181t^4$	$7.0282409 - 10$ $4.5520763 - 10$ $2.1746202 - 10$ $9.9303494 - 20$

142. Critical Temperature.—Computations made by the foregoing formulæ (§ 141) show that the internal latent heat of vapors (r_1) decreases as the temperature rises, since the terms containing the temperature as a factor act to diminish the value of the constant for each vapor. There must be a temperature therefore for each vapor at which r_1 will become zero for that vapor, or, in other words, a temperature at which there is no internal work done in disgregating the particles of the medium, but at which it behaves as a perfect gas. The distinction between the liquid and its vapor has disappeared, and it would be correctly inferred that above this temperature, pressure alone—without a concurrent lowering of temperature—would not liquefy the vapor or gas. This temperature at which the value for r_1 or the internal latent

heat disappears has been called the critical temperature for any vapor; and it has been wisely advocated to use the term gas for a vapor above its critical temperature, while below that temperature it is a vapor.

The accepted determinations so far have been those of Avenarius (Poggendorff's Annalen, vol. 151, 1874), in which volatile liquids were heated in strong glass sealed tubes until the liquid all disappears and the tube is full of gas. His experiments, compared with the computations based on Regnault's experiments from equations deduced for the internal latent heat, give the following:

TABLE OF CRITICAL TEMPERATURES.

	Experiment.	Calculation.
Ether...................	196.2 C.	196.8 C.
CS$_2$...	276.1	274
CCl$_4$..	292.5	298.7
Aceton.........	246.1	230.4

The critical temperature for water as calculated from Zeuner's formula is 720° C. (1328° F.), which is beyond the present limits of experiment.

143. Increase of Entropy of a Liquid and its Vapor.— It is usual in considering liquids with their vapors to treat of a unit of weight, from which a portion x is raised from liquid to vapor. Hence $1 - x$ is the weight of liquid, and $1 \times x = x$ is the weight of vapor. If such unit of weight of liquid is raised from freezing-point to a temperature t, and the part x is evaporated at that temperature, the increase of entropy will take place in two stages, and the total increase will be their sum. The heat of the liquid portion is determinable from the Regnault experiments and data given in § 139 as denoted by q, so that when t denotes the range above centigrade 0°, the increase in entropy from that at zero represented differentially by the symbol $\dfrac{dq}{T}$ can be written:

$$\theta = \int_0^t \frac{dq}{T} = \int_0^t \frac{cdt}{T} = c \text{ hyp. log } \frac{T}{T_0},$$

as foreshadowed in § 124. In this expression c is the specific heat, considered as constant for the range in question. If it is not constant, as, for instance, was made clear by Regnault in the case with water (§ 139), the entropy of the liquid is to be calculated in steps or stages from the zero-point to that desired. Referring to Regnault's experiments, the specific heat between 0° and 5° C. is 1.0072; from 5° to 10°, 1.0044; from 10° to 15°, 1.0016. This would make the entropy for a temperature between 10° and 15°—say at 13°, or 65° F., to be made up of

$$1.0072 \text{ hyp. log } \frac{T_5}{T_0} + 1.0044 \text{ hyp. log } \frac{T_{10}}{T_5} + 1.0016 \text{ hyp. log } \frac{T_{13}}{T_{10}}$$
$$= 0.04663.$$

For other liquids the data of the tables in § 139 can be used which give values for q directly. By differentiating these equations, the resulting equation has the form $\frac{dq}{dt}$, which gives the specific heat at the desired temperature t, since $\frac{dq}{dt} = c$ from the derivation of equations giving values for q, as well as by definition of the terms used. So that, if ether be taken for which (§ 139)

$$q = 0.52901t + 0.0002959t^2;$$

then

$$\theta = \int \frac{dq}{T} = \int_0^t (0.52901 + .0005919t) \frac{dt}{T},$$

which becomes

$$\theta = (0.52901 + .0005919t) \text{ hyp. log } T.$$

144. Increase of the Entropy of the Vapor.—In the case of a vapor which is formed from a liquid with which it is in contact, the heat is continually added at the constant temperature at which the vapor stands or is formed. Hence the weight x of the original unit weight receives a quantity of heat q', which is represented by the heat of vaporization r, which is the difference between the total heat λ and the heat of the liquid q (§§ 138, 139, and 140). Hence it follows that the increase in entropy for the weight x will be represented by

$$\frac{xr}{T}.$$

The entire increase in entropy will be therefore the sum of

$$\theta + \frac{xr}{T} = \frac{xr}{T} + \int_0^t \frac{cdt}{T},$$

as was explained in § 124.

The convenient approximation is also to be remembered in the absence of tables or formulæ for θ, the entropy of the liquid, whereby the specific heat is assumed to be constant; so that

$$\theta_1 - \theta_2 = c \int \frac{dt}{T} = c \text{ hyp. log } \frac{T_1}{T_2};$$

whence for adiabatic relations, and $\phi - \phi_2 = 0$,

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{xr}{T} + \theta$$

can be written

$$\frac{x_2 r_2}{T_2} = \frac{x_1 r_1}{T_1} + c \text{ hyp. log } \frac{T_1}{T_2},$$

in which x_2 is the unknown quantity for which the equation is to be solved.

145. Superheated Steam, Total Heat of.—Superheated steam has been already defined (§ 119). The total heat of such superheated steam is therefore that due to the increase of its temperature considered as a gas by the addition of the sensible degrees of temperature above that total heat which it had as a saturated vapor, as given in § 138. The formula for the total heat, when the sensible temperature of saturation at any pressure is t and the superheat temperature is t_2 becomes for Fahrenheit units:

$$\lambda = 1091.7 + 0.305(t - 32) + c(t_2 \cdot \cdot t).$$

The value of c, or the specific heat of steam becomes at once of significance.

146. Specific Heat of Steam.—It is obvious that there will be a difference in the specific heat of steam at constant pressure and at constant volume, as is the case with all gaseous media, but furthermore it would be expected that there would be a different specific heat for constant pressure at or near the saturation-point from that prevailing when the steam is superheated to a degree permitting considerable cooling before condensation to liquid began.

Regnault's accepted value for the specific heat of superheated steam at constant pressure is

$$c_p = 0.4805,$$

which is the mean of three determinations, giving

$$0.48111,$$
$$0.47963,$$
$$0.48080.$$

The specific heat of saturated steam has to be more definitely defined to secure exactness, by stating that it is the

quantity of heat which must be added to one unit o weight of steam, when the temperature is raised one degree and the pressure increased the corresponding amount while the steam remains dry and saturated under this change. It will appear, therefore, that the specific heat of a dry and saturated vapor is not exactly like either of the others, as the condition of saturation imposed compels a change of both pressure and volume to maintain the imposed equilibrium. The accepted statement for the value of the specific heat of saturated steam has therefore the form

$$C = 0.305 - \frac{r}{T}.$$

147. Negative Specific Heat of Saturated Steam.— If the foregoing equation be solved for C, by means of the relations $r = \lambda - q$, and the values for λ and q already given (§§ 138 and 139) for several temperatures, it will result that within usual limits of practice the specific heat C comes out negative. For example:

$$
\begin{aligned}
&\text{At} \quad 0° \text{ C. or } \ \ 32° \text{ Fahr.,} \quad C = -1.911 \\
&\text{``} \quad 50 \text{ `` `` } 100 \quad \text{``} \quad \ \ C = -1.461 \\
&\text{``} \quad 100 \text{ `` `` } 212 \quad \text{``} \quad \ \ C = -1.131 \\
&\text{``} \quad 150 \text{ `` `` } 300 \quad \text{``} \quad \ \ C = -0.879 \\
&\text{``} \quad 200 \text{ `` `` } 392 \quad \text{``} \quad \ \ C = -0.676
\end{aligned}
$$

It will be apparent that the value for C is approaching a point of inversion at which it will be zero and beyond which it will become positive, as for many of the other heat media, such as ether. This point, however, is beyond present experimental knowledge. The negative value within experience means that when temperature and pressure are increased together, the steam will become superheated unless heat is abstracted by doing work. Or, in other words, a sudden expansion or drop of pressure, if accompanied also with a

drop of temperature, causes a part of the steam to be con-
densed. The heat freed by the condensation of that weight
which is condensed serves to keep the uncondensed part in
the required state·of saturation under the changed conditions.
This was shown experimentally by Hirn (Bulletin de la So-
ciété Industrielle de Mulhouse, tome 133). This is quite
different from the condition discussed in § 131, in which the
temperature was assumed to be maintained at the point be-
longing to the higher pressure, while the pressure fell. The
negative specific heat of steam acts in an unjacketed and
conducting cylinder to increase the cylinder condensation,
since the pressure is falling and heat is being withdrawn by
radiation outwardly, while the piston-work is being done at
the expense of the heat brought into the cylinder.

**148. Specific Heat of Superheated Steam at Constant
Volume.**—The specific heat of superheated steam at constant
volume can be calculated by means of a ratio between the
two specific heats, assumed by Zeuner to be $\frac{4}{3}$ or 1.333,
whence for various pressures in pounds per square inch

	5	50	100	200	300
$C_v =$	0.351	0.348	0.346	0.344	0.341

or is apparent as a variable.

149. Specific Volume of Superheated Steam.—The vol-
ume of superheated steam in metric equivalents and centi-
grade degrees as given in the following table presents the
results of experiments by Hirn. The comparison column is
deduced from an equation expressed in kilograms per square
meter for the volume belonging to any pressure and temper-
ature T, which has the form in

$$\text{Metric units.... } pv = 51.3T - 188p^{\frac{1}{4}};$$

and in   English "   ... $pv = 93.5T - 971p^{\frac{1}{4}}.$

When the same equation is applied near the limit or near
the condition of saturation it seems to apply fairly well.

SPECIFIC VOLUME OF SUPERHEATED STEAM.

Pressure in Atmosphere.	Temperature Centigrade.	Cubic Meters per Kilogram.	
		Hirn's Experiments.	Calculated Values.
1	118.5	1.74	1.75
1	141	1.85	1.87
3	200	0.697	0.699
4	165	0.4822	0.476
4	200	0.522	0.520
4	246	0.5752	0.577
5	162.5	0.3758	0.376
5	205	0.414	0.418

150. Specific Volume of Saturated Steam.—The volume occupied by a unit weight of saturated steam has already been given in the steam-tables presented in § 136. It will be of significance to refer to experiments to determine these values. Those of Fairbairn and Tate in England (*Philosophical Transactions*, vol. 150, p. 185, 1860) are among the most complete. Fig. 52 represents a diagram **to illustrate** their method. *A* and *B* are globular vessels containing slightly varying weights of water. They are connected by a tube containing mercury, and the whole apparatus is immersed in a bath whereby any desired temperature may be given to the globular vessels and their contents. As long as any water is in either

FIG. 52.

globe the vapor in each will have the same tension, and the mercury will stand even in the two legs. There will be a temperature, however, at which the water in that vessel which has originally the least weight of water in it will become altogether vapor, in a dry and saturated state, and, if this point can be accurately observed, the density can be found from the known weight and observed volume. Beyond this point of dryness and saturation the steam in a state of complete vaporization becomes superheated by addition
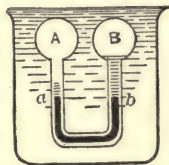
of heat, and the mercury will rise in the dry leg. The pressure for saturated steam at any temperature is greater than for superheated steam at that same temperature. The difficulty and uncertainty in the experiment is due to the chances of error and uncertainty in fixing on the period of complete vaporization and the corresponding temperature.

The Tate and Fairbairn formula which they deduced to represent their work is of the form

$$V = 25.62 + \frac{495 \cdot 13}{P + 0.72}.$$

The term V is the volume of steam compared to that of the water which produced it, and P is the pressure in inches of mercury.

This transformed to cubic feet per pound, and with p in pounds per square inch, becomes

$$V_1 = 0.016 \left[25.62 + \frac{24317.5}{P_1 + 0.365} \right].$$

Zeuner's empirical formula is

$$D = a p^{\frac{1}{n}},$$

in which $a = 0.6061$ and $\frac{1}{n} = 0.9393$. D is in kilograms per cubic meter, and p is in atmospheres.

151. Condensation in Adiabatic Expansion of Steam.— It will be apparent in an adiabatic expansion in which by definition the external work is being done at the expense of the intrinsic heat energy of the steam, that with the use of superheated steam a certain amount of expansion will occur before the steam becomes saturated, and that if the expansion be carried further the steam will become moist by condensation of a part of it. The equations of § 144 when applied to an actual case with the proper data will make this clear. Let a unit weight of steam ($x_1 = 1$) be taken at 100

pounds pressure above vacuum and let it be superheated to a temperature of 400° F.: then let it be expanded adiabatically down to atmospheric pressure. Then in the formula

$$\frac{r_1}{T_1} + \theta_1 + 0.4805 \text{ hyp. log } \frac{T_s}{T_1} = \frac{x_2 r_2}{T_2} + \theta_2;$$

all data are known or to be taken from tables except x_2, the final state of steam in the same unit of weight. When, therefore,

$$\theta_1 = 0.4733; \qquad t_1 = 327.6; \qquad r_1 = 884;$$
$$\theta_2 = 0.3143; \qquad t_2 = 213.9; \qquad r_2 = 965.1;$$

the equation becomes

$$\frac{884}{788.3} + 0.4733 + 0.4805 \text{ hyp. log } \frac{860.7}{788.3} = \frac{965.1 x_2}{673.7} + 0.3143,$$

and there will result

$$x_2 = 0.923,$$

or a little less than 8 per cent of the steam has become water, in the process of expansion, in spite of the initial superheat.

If instead of T_s, or the absolute superheat temperature, the temperature of saturation had been used, the formula and calculation would show the amount of condensation to be expected from steam which is simply dry. See also Chapter XIV, § 200 *et seq.*

152. Evaporation from a Feed-water Temperature.— The formulæ for total heat of steam (§ 138) give the total heat counted from freezing-point of water. When the feed-water which is pumped into the boiler of a steam-engine is at some higher temperature than this, it is apparent that the coal burned in the boiler-furnace does not have to supply as much heat per pound of water evaporated as the formula would indicate. The feed-water being assumed at t' F., the

total heat required to make steam at t degrees temperature becomes for each pound of water evaporated

$$Q_1 = 1091.7 + 0.305\,(t - 32) - c(t' - 32),$$

or (if the specific be called unity between t and $32°$ F., instead of using its more exact value)

$$Q_1' = 1091.7 + 0.305\,(t - 32) - (t' - 32).$$

153. Evaporation from and at 212° F.—A very convenient, practical, and useful deduction is made from the formulæ for total heat of steam (§§ 138–144) to compare the evaporative performances of steam-boilers working under different pressures. It is apparent that more heat goes into each pound of water at the higher pressures. Hence it has been agreed to reduce all conditions of feed-water temperature and evaporative temperature and pressure to the condition of a feed-water temperature of 212°, and an evaporation of the water at that temperature into steam at atmospheric pressure, with a temperature of 212°. This has been shortened into the compact expression, "Evaporation from and at 212°."

The pressure of one atmosphere is 14.7 pounds per square inch, and at 212° F., which is the corresponding boiling temperature, the heat necessary to make water at that temperature into steam at that pressure is

$$965.7 = 966 \text{ B.T.U.}$$

If, then, from the preceding paragraph, the total heat Q required to vaporize a weight of water W be observed from a test, in which the feed-water was introduced at t' and the evaporation took place into steam at t, the total heat which went into the evaporated water was the product

$$QW.$$

If the evaporation had taken place from and at 212°, Q would have been 966 for each pound, so that

$$966x$$

would have been the equivalent heat absorption if x is the corresponding weight of water evaporated at that atmospheric pressure. Equating these,

$$QW = 966x \quad \text{or} \quad x = \frac{QW}{966},$$

gives the pounds of water which would have been evaporated from and at 212°. This may be either the total equivalent evaporation, or the equivalent evaporation per pound of coal, or per horse-power per hour, according to the unit used for W.

154. Rankine's Factor of Evaporation.—Rankine computed a table giving values for a factor,

$$F = \frac{Q}{966},$$

wherewith to multiply the quantity W to produce the desired weight X (Rankine, "Steam-engine and other Prime Movers," pp. 255, 256). His equation for F is

$$F = 1 + \frac{0.3(t_1 - 212°) + (212 - t_2)}{966},$$

in which t_2 is the temperature of the feed-water, and t_1 the temperature at which vaporization actually took place.

155. Theoretical Evaporation of Water per Pound of Fuel.—The formula for total heat of steam under various conditions renders it possible to predict the limits of evaporative capacity with any fuel whose calorific power is known or assumed. The pounds of water evaporated per pound of fuel burned will be the same as the quotient found by dividing the total heat at that temperature into the calorific power

(§§ 22, 58, 59).   For example, while with pure carbon, with
a calorific power of 14,500 heat-units, the evaporation from
a feed-water temperature at 212° will be

$$\frac{14500}{966} = 15$$

pounds of water per pound of carbon, as was already shown in
§ 59, at higher pressures, or with cooler feed-water tempera-
tures, the value of the divisor increases while the dividend re-
mains constant, and hence the quotient grows smaller.   The
effect of this is to make the maximum theoretical evaporation
per pound of fuel burned less as the pressures increase at
which the steam is formed.

156. Output of a Steam-boiler in Heat-units. — The
product of the pounds of water evaporated by a boiler into the
total heat of the steam at that pressure, as determined from
the foregoing formulæ, gives the heat-units which that boiler
is delivering.   These may be expressed in any unit, per hour,
or per boiler horse-power, or per day, or per 1000 pounds of
steam delivered to engine, or in any form.   The accepted
boiler horse-power unit of 30 pounds of water evaporated into
steam at 70 pounds pressure from a feed-water temperature
of 100° is equivalent to an evaporation of 34.488 pounds of
water from and at 212°.   The product,

$$34.488 \times 965.7 = 33305 \text{ heat-units},$$

is the heat-units per boiler horse-power from and at 212°, ac-
cording to the standard of 1885 of the American Society of
Mechanical Engineers.   Any other similar computation can
be made from the observed results of a boiler-test.

157. Efficiency of a Steam-boiler.—The efficiency of a
steam-boiler as an appliance for getting the heat-units avail-
able in the fuel into the heat-carrier or medium whereby they
are to be utilized would appear to be the relation between
the heat-units in the output as compared with the total heat-

units charged into the furnace. (See § 59.) That is, if the efficiency were unity, or one hundred per cent., each pound of fuel should evaporate a weight of water W which, multiplied by its total heat in heat-units, should give a product equal to its calorific power; or,

$$\text{calorific power} = W \times Q.$$

This is the statement of § 155 reversed. The actual evaporation, however, is not W but a smaller weight W', raised to the same total heat, Q. Hence for any boiler,

$$\text{efficiency} = \frac{W' \times Q}{W \times Q} = \frac{W'}{W} = \frac{\text{actual output in heat-units}}{\text{calorific power of the fuel}}$$

taken per hour, or per horse-power, or per pound, as may be convenient.

A comparison of actual and theoretical output in tests of modern economical boilers will show that it may claim to be a fairly efficient apparatus for its purpose.

CHAPTER XII.

WORK DONE BY ELASTIC HEAT MEDIA IN HEAT-ENGINES. CYLINDER DESIGN.

160. Introductory.—In the foregoing chapters the subject of heat has been discussed first from the standpoint of its generation or liberation from a fuel or combustible; and, secondly, with respect to the effects which heat produces upon a suitable medium to convey the energy of that heat to the organ or motor which is to utilize it. This chapter will discuss the action of such heat-medium exerting an effort by reason of the elastic tension which has been imparted to it by heat to produce motion of a suitable piston in a cylinder and overcome a resistance expressed in pounds exerted through a space expressed in feet.

Attention was called to the general truth in Chapter II, § 7, that a work in foot-pounds could be represented by an area equivalent to that of a rectangle whose height or altitude was proportional to pounds on any accepted scale and whose length or base was proportional to feet. The product of the base and altitude would therefore be a product expressed in foot-pounds. It will further be recalled that for a piston-engine which has a piston-area A and a length L, the work in any time in which N strokes are made in such cylinder while a pressure P is exerted over the area A will be a product in foot-pounds represented by

$$W = PA \times LN,$$

which becomes expressed in horse-power per minute by making N represent the number of traverses of the piston per minute, and dividing both members by 33,000, or

$$HP = \frac{PALN}{33000}.$$

It becomes necessary now, however, to take up this work-formula in more exact detail for a fuller discussion. The thermal lines discussed in §§ 119–125 now have a significance.

161. Work Done with Constant Pressure in the Cylinder.—Since the product AL is equivalent to the volume of the cylinder filled once, and ALN is the volume filled during N traverses or $\frac{N}{2}$ revolutions if the engine is double-acting, it becomes apparent that AL becomes V for one traverse, and ALN is V for N traverses. It is possible, therefore, to write

$$W = PLA = PV$$

for one traverse, provided it be clearly kept in mind that P and A must be kept in the same unit. This is most easily done by expressing P in pounds per square foot, and A in square feet. It is just as true to use pressure in pounds per square inch p, which is equal to $\frac{P}{144}$, provided the area a is in square inches, and $a = 144A$. Then

$$PLA = \frac{R \times 144A \times L}{144} = paL.$$

The length L must remain in feet, in any case, which makes this latter method liable to confuse. But it is in the form

$$\text{Work} = PV$$

that the formulæ applicable to heat-engines have so far appeared.

The resistance against which the heat-motor works is supposed to be a uniform effort, expressed in pounds to be overcome through the given path. The discussion of the flywheel as a regulator or accumulator to compensate and control irregularities of the resistance is aside from the present purpose. The resistance will be assumed to be able to be kept constant.

Such resistance can therefore be represented by a rectangle, $ABCD$ (Fig. 53), in which the horizontal is the space s, through which the resistance is overcome in one traverse of the motor-piston, and the vertical or height is proportional to the intensity f of that resistance in pounds. In order that
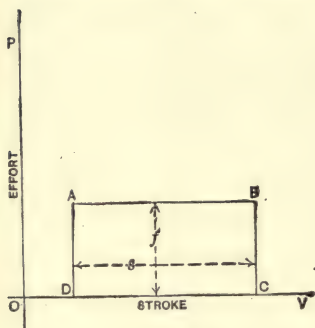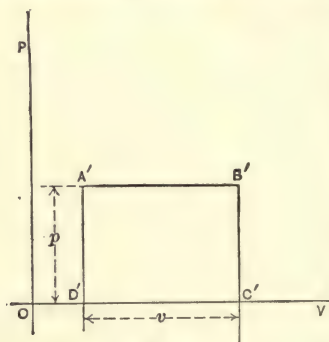
Fig. 53.　　　　　Fig. 54.

this constant resistance may be overcome, the cylinder of the motor must generate an equal product or area in which the horizontal shall be the volume filled by the elastic heat-medium, and the vertical is proportional to the pressure P. That is, the work in foot-pounds $W = fs = ABCD$ of Fig. 53 must be equal to the area $PV = A'B'C'D'$ in Fig. 54.

It is to be observed that there is an infinite number of rectangles or other figures which have the area $PV = fs$. The two need not have the same altitude or base, provided only their areas are the same. This represents the simplest case which occurs in tank-pumping with small pumps without fly-

wheel, where the parts are so light as to permit their inertia to be neglected, and where no excess of energy is required from the motor to cause the engine to pass its centres at the ends of each stroke so as to reverse the motion. The effort of the elastic medium is a constant effort of the same intensity and the volume generated is that which exists at the end of the stroke. Instead of using the stroke as the unit, the work of a minute might have been chosen involving N strokes. Instead of either of these, the volume might have been made that occupied by a pound weight of the medium at the constant pressure P, and the corresponding value of fs calculated back from the value of the corresponding PV. In any case one or the other of the two factors must be assumed, and the value of the other calculated to meet the case.

162. Constant Pressure-work with Air or Permanent Gases.—The condition presented by the rectangular area for PV is not considered a desirable or economical one. The difficulty is caused by the fact that at the completion of the working-stroke the contents of the cylinder must be voided to permit the piston to return to its original position against the least possible internal resistance in the cylinder itself. Hence a volume of medium V at a pressure or tension P would be wasted at each stroke. In the case of air, if it be assumed that no temperature changes occur, and the work represented by PV was put into the working medium by the doing upon it of an external work by a compressor which is equal to that same PV, then the compressed air acts exactly like incompressible water or like a solid mass to transmit the work of the compressing motor to the air-motor cylinder. A certain weight or volume of air is displaced mechanically, and in the strictest sense the transmission is not a heat-engine transaction at all. Usually, however, temperature changes do come in.

The work represented in bringing up its tension from atmospheric pressure to its working tension is thrown away at

the exhaust, because the potential energy is not exerted which it might have given out in expanding down to atmospheric pressure again. The drop in pressure takes place outside of the cylinder. The heat expended in raising the medium to the higher pressure will be a matter of later discussion.

163. Constant Pressure-work with Steam.—In the case in which a steam-cylinder receives steam through a pipe from a boiler during the entire stroke of the piston, the engine is said to work without expansion, or non-expansively (Fig. 1). This is true in the sense that the terminal pressure and temperature of the steam at the moment when exhaust begins are the same as at the beginning, and all the heat and potential energy in the weight of steam which fills the final volume of the cylinder is voided during the exhaust. What occurs in this case is an expansion in the boiler and not in the engine. The fire or source of heat has to furnish the amount of heat represented by the total heat of steam at the pressure p_1 at which it works in the cylinder for each pound of steam represented in the terminal volume v_1. The data of §§ 136 and 138 enable this calculation to be made. The rejection of so much potential energy at the end of the working-stroke, which has been imparted to the medium by the heat of the fire, and which might be utilized for the doing of external work in the engine, make this method of working less economical and efficient than methods now to be discussed.

164. Work Done by an Elastic Heat-carrier Expanding in a Cylinder. Cut-off or Degree of Expansion.— It has already been made apparent (§ 112 *et seq.*, Chapter X) that the elastic tension of a heat medium is a function of its temperature or is dependent upon it. The necessity for disposing of the working-volume of the medium at the end of the stroke, and its rejection from the heat-engine proper, would at once suggest the advisability (if it were possible to secure this result) of making the elastic heat-carrier surrender

all the heat energy which could be gotten from it before it
leaves the cylinder. This implies the maximum lowering of
its temperature and pressure, which is wise to demand, *within
the cylinder itself*, and after the disconnection of the required
weight of medium from its source of heat. Hence the prin-
ciple is availed of which is expressed mathematically by the
equation,

$$p_1 v_1 = pv = RT$$

for isothermal expansion; or the expression,

$$p_1 v_1^n = pv^n = \text{constant}$$

for an adiabatic expansion. As applied in a practical way,
this means that if the elastic heat-carrier be admitted from
the heating-appliance to fill a small volume v_1, and then that
volume is increased up to the final volume represented by the
entire piston-displacement in one stroke (or v), the final or
terminal pressure will be as much lower than the initial pres-
sure as the final volume was greater than that filled by the
medium while it was flowing into the cylinder from the out-
side reservoir of pressure. With the non-condensable media,
it is therefore desirable that the terminal pressure should be
that of the atmosphere surrounding the cylinder if the ex-
haust takes place in the open air, unless some practical con-
siderations should give another consideration greater weight,
which is hereafter to be discussed. In condensing engines
(Chapter IX) a similar theory would demand that the terminal
pressure might be that of the vessel into which the exhaust
takes place—which latter expression can be made generally
applicable, subject only to the limitations to be hereafter im-
posed. The heat-carrier thus carries out of the cylinder the
minimum amount of available energy.

This principle of working the heat medium so as to make
it expand to a larger volume and lower pressure, and do
work against the resistance in thus expanding, is secured in

practice by cutting off the admission of the medium at a designated point in the stroke of the piston. The point where admission of heat-medium and heat-units to the working-cylinder ceases is called the point of cut-off, and is expressed as a fraction of the entire piston-stroke as unity. In a cylinder of uniform section the volume filled up to the point of cut-off of the admission will bear the same relation to the final volume at the end of the stroke as the ratio of the lengths of the stroke bear to each other. Volumes and lengths can therefore be used indifferently.

The final volume of the elastic medium will be to the volume present when cut-off took place in a ratio which is the reciprocal of the fraction expressing the point of cut-off. That is, the degree of expansion, or the ratio of expansion, will be 2 for a cut-off at one half stroke, since the final volume will be twice that present when cut-off occurred. This distinction must be carefully observed lest confusion occur. In using cut-off terms, the final volume is the denominator and the cut-off volume is the numerator; in handling the ratio of expansion or the number of expansions which is usually designated by r and is a number greater than unity, the final volume is the numerator and the cut-off volume the denominator.

In handling expansions in non-conducting cylinders, where the working medium gets no heat from outside, but expends its own heat energy in doing work with the piston, its temperature may drop inconveniently in expanding down to the pressure prevailing in the space into which it exhausts. Furthermore, it will be apparent that a lowering of the final pressure must be accompanied with a less amount of forward effort at the end of the stroke than at the beginning, and hence a larger piston-area is required than for the constant value of the pressure. This means a bulkier and probably heavier engine, and more fly-wheel mass to compensate the irregular effort. It will be seen, therefore, that it may not be desirable

to carry the lowering of pressure to the practicable limit in actual cases. This further opens the question of the considerable potential energy rejected from a heat-motor with its rejected medium, because of the impossibility of capturing it for use. These will be referred to later in a different connection.

165. Work of a *pv* Diagram Represented by an Area. —It has been already said that the work of a piston-engine could be represented by the area of a diagram (§ 7). Let Fig. 61 represent such a *pv* diagram, in which v_1 is the
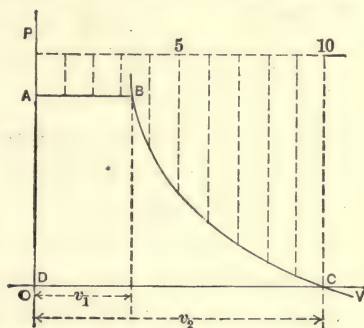
FIG. 61.

volume filled at cut-off and v_2 is the final volume, while the vertical ordinates reproduce the pressures prevailing at each point of the piston-stroke. If this is a *pv* diagram from an actual engine *v* while variable is yet known, and the problem resolves itself into finding a value for the varying *p* which shall be a mean value for that stroke, whereby the effort of the medium can be equated to the mean value of the resistance during the same period. The inequalities of the varying value for *p* *in each stroke* can be provided for by the flywheel, and must be so taken care of. If the mean value of *p* is too small, the engine will slow down until the diminished distance through which the resistance is moved restores equilibrium. If *p* is too large for the mean resistance, the engine accelerates its speed until the greater space per min-

ute for the resistance balances the excess of effort, or the internal resistances of the motor itself supply the excess of resistance.   It is from the governor of the engine that a proper control of the mean effort must be expected.

The area of the *pv* diagram, as determined by observation or otherwise as to its bounding curves, will give a figure expressing a number of square inches or square feet.   If this number be divided by the measured length, the quotient will be the height of an equivalent rectangle having the same length.   This value for the mean pressure (called *p*) can be substituted in the formula,

$$HP = \frac{p_m LAN}{33000},$$

in which A, L, and N are taken from the actual case.

The area of the *pv* diagram can be found by dividing it lengthwise into conveniently short portions, and taking the area of each portion and making a summation of the fractional areas.   If the number of areas be made ten, their bases being of equal length, the sum of their mean heights divided by ten (displacing the decimal point one place to the left), gives the mean height of the diagram as a whole.   If a planimeter is at hand, the area is determined directly within the error of that instrument.

To apply Simpson's rule for determining an area the diagram is divided vertically by n ordinates.   The first one is called p_o and the last one p_n.   Then the area A is given by the formula:

$$A = \frac{1}{n} l \left(\frac{p_0 + p_n}{2} + p_1 + p_2 \cdots \cdot p_{n-1} \right),$$

when l is the measured length.   Dividing this area by the length l, the mean pressure p results; or,

$$\frac{A}{l} = pm = \frac{1}{n} \left(\frac{p_0 + p_n}{2} + p_1 + p_2 \cdots p_{n-1} \right).$$

If the expulsion of the working medium used in the pre-
ceding stroke exerts a negative pressure upon the piston on
its working stroke, then the pressure ordinates cannot be
measured from a zero line of no pressure. The working dia-
gram representing the effective pv area is displaced up-
ward above the horizontal reference line of zero pressure
(Fig. 62). The value of that constant back pressure p_3 must
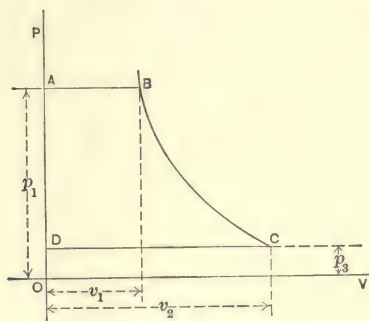
FIG. 62.

be subtracted from the mean forward pressure p_m, making the
effective pressure $p_e = p_m - p_3$. Or the subtraction may be
made by scaling the values for p_0 to p_n from the back-pressure
line instead of the zero-pressure line. The planimeter also
gives p_e directly.

It is apparent that the steam-engine indicator draws a
pv diagram from which the mean effective pressure usually
designated M.E.P. is one of the primary deductions. A
calibrated spring equilibrating the pressure in the cylinder
against a piston of known area can be made to compel a
pencil or tracing-point to draw a diagram giving the values
for p and p_3 at each point of the stroke of the engine-piston.

**166. Work of an Elastic Heat Medium Expanding Iso-
thermally.**—The foregoing discussion applies to any pv dia-
gram bounded by any curves. If the special case be assumed
of an expansion according to the isothermal law in a cylinder
of highly conducting material through which heat from outside

may be supplied to the working medium as it requires it in expanding, certain simple rules of procedure derived from the calculus enable the area of the pv diagram or the mean pressure p_m to be foretold when the value of the initial pressure is given, and the ratio $\dfrac{v_1}{v_2}$.

In Fig. 63 the total work represented by the diagram is the sum of the areas of the rectangle $ABCD$ equal to p_1v_1, and the curved portion BEC bounded on its upper side by an
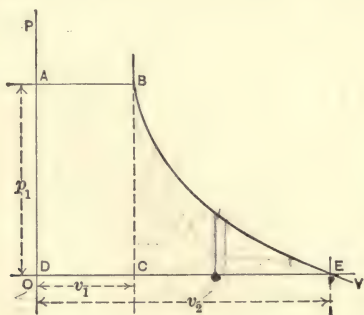
Fig. 63.

equilateral hyperbola BE, if the expansion-curve is an isothermal curve in which $p_1v_1 = p_2v_2$. The work W_1 of the rectangular part is obviously p_1v_1. The work W_2 of the hyperbolic part will be expressed by the differential equation,

$$W_2 = \int_{v_2}^{v_1} p\,dv = p_1v_1 \int_{v_2}^{v_1} \frac{dv}{v_1},$$

since $p_1v_1 = pv$, and p_1v_1 is not a variable.

The integration of that expression by the methods of the calculus between the limits v_1 and v_2 gives

$$W_2 = p_1v_1 \text{ hyp. log } v_2 - p_1v_1 \text{ hyp. log } v_1;$$

or

$$W_2 = p_1v_1 \text{ hyp. log } \frac{v_2}{v_1}.$$

The total work is the sum of the two portions:

$$W_1 + W_2 = p_1 v_1 + p_1 v_1 \text{ hyp. log } \frac{v_2}{v_1}$$

$$= p_1 v_1 (1 + \text{ hyp. log } r),$$

since the ratio of expansion r is the same as $\dfrac{v_2}{v_1}$.

If the foregoing expression for work is the area of the pv diagram, the mean value for p_m will be found by dividing through both members by v_2, the known length of the stroke or final volume of the cylinder. Hence

$$p_m = \frac{W}{v_2} = \frac{p_1 v_1}{v_2}(1 + \text{ hyp. log } r).$$

But

$$\frac{v_1}{v_2} = \frac{1}{r}.$$

Hence, the mean pressure becomes

$$p_m = p_1\left(\frac{1 + \text{ hyp. log } r}{r}\right),$$

from which the pressure to be inserted in the horse-power formula can be calculated from any assumed initial pressure in boiler or pressure reservoir, when the effects of clearance and compression are not considered, and the expansion is isothermal. If there is a back pressure p_3 it must be subtracted from p_m before the latter is inserted in the horse-power formula,

$$\text{H.P.} = \frac{p_e L A N}{33000},$$

because p_m is the gross effective mean pressure counted from lines of zero pressure.

167. Work of an Elastic Heat Medium Expanding Adiabatically.—In the foregoing paragraph, where the final

state of the medium as respects heat energy was the same as the initial state, by the supply of the necessary heat to the expanding medium through a conducting cylinder wall by a hot jacket or otherwise, the external work was all done by the applied heat. This is obvious for the admission stage $p_1 v_1$, and is made clear for the expansion stage by considering that the final pressure and temperature are no less than they would be if the piston had been moved from outside by some force, whereby the medium expanded doing no work. Any work done by expanding without such addition of heat from without must be accompanied by a fall of the heat energy represented by the work done during such expansion. If the cylinder is a non-conducting and non-absorbing one for heat, and work is done during expansion, the terminal pressure ought to be less at the end of such adiabatic expansion than when the law $pv = RT$ is true. Hence the equation of the form

$$p_1 v_1^n = p_2 v_2^n$$

will be true, if n has a proper value, greater than unity. Rankine ("Steam-engine," p. 392) gives for the exponent a value $\frac{10}{9}$ when the initial pressure is not less than one atmosphere nor more than twelve atmospheres, in a non-conducting cylinder. For a jacketed engine with pressure between 30 and 120 pounds initial pressure, and for ratios of expansion between 4 and 16, he gives a value of $\frac{17}{16}$ for the exponent. Zeuner considers 1.0646 nearer than Rankine's value of 1.111.

The most general form of the work-equation is conveniently applied in this case,

$$\text{Work} = \begin{bmatrix} \text{admission} \\ \text{work} \end{bmatrix} + \begin{bmatrix} \text{expansion} \\ \text{work} \end{bmatrix} - \begin{bmatrix} \text{back-pressure} \\ \text{work} \end{bmatrix}$$

$$W = \quad p_1 v_1 \quad + \int_{v_1}^{v_2} p\, dv \quad - \quad p_2 v_2 \,;$$

but since $pv^n = p_1v_1{}^n$, the integration of the expansion work becomes

$$W_1 = p_1v_1 \int_{v_1}^{v_2} \frac{dv}{v^n} = \frac{p_1v_1 - p_2v_2}{n-1},$$

which can be transformed into

$$W_1 = \frac{p_1v_1}{n-1}\left[1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right],$$

since

$$p_1v_1{}^n = p_1v_1(v_1)^{n-1} = p_2v_2{}^n = p_2v_2(v_2)^{n-1};$$

whence

$$p_2v_2 = \frac{p_1v_1(v_1)^{n-1}}{(v_2)^{n-1}} = p_1v_1\left(\frac{v_1}{v_2}\right)^{n-1}.$$

This factors into the expression

$$W = \frac{p_1v_1}{n-1}\left[n - \left(\frac{v_1}{v_2}\right)^{n-1}\right],$$

when p_1v_1 be added in the form of

$$p_1v_1 = \frac{n-1}{n-1}(p_1v_1).$$

As before, the mean pressure is $\dfrac{W}{v_2}$, or

$$p_m = \frac{W}{v_2} = \frac{\dfrac{p_1v_1}{n-1}\left[n - \left(\dfrac{v_1}{v_2}\right)^{n-1}\right]}{v_2};$$

or, if $r = \dfrac{v_2}{v_1} =$ ratio of expansion, then

$$p_m = \frac{p_1}{n-1}\left(\frac{n}{r} - \frac{1}{r^n}\right),$$

since

$$\frac{v_1}{v_2} = \frac{1}{r} \quad \text{and} \quad \left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{1}{r}\right)^{n-1},$$

and

$$p_1 \frac{v_1}{v_2} \left(\frac{v}{v_2}\right)^{n-1} = p_1 \frac{1}{r^n}.$$

The expression for mean pressure can be substituted for p_m in the horse-power formula for the work when expansion is adiabatic, and the desired work per minute or per stroke worked out as before, when the value for n is assumed or is known. When Rankine's value $\frac{10}{9}$ is used for n, then

$$p_m = 9p_1\left(\frac{10}{9r} - \frac{1}{r^{\frac{10}{9}}}\right) = p_1\left(\frac{10}{r} - \frac{9}{r^{\frac{10}{9}}}\right).$$

168. Adiabatic Work in Terms of Pressures.—Since

$$p_1 v_1^n = p_2 v_2^n, \quad \text{then} \quad \frac{p_2}{p_1} = \frac{v_1^n}{v_2^n}$$

whence by extracting the nth root

$$\frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}},$$

and by raising both members to the $n - 1$ power,

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

Hence the equation for work of expansion of the preceding paragraph becomes

$$W_1 = \frac{p_1 v_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}};\right.$$

and adding the admission work p_1v_1,

$$W = p_1v_1 + \frac{p_1v_1}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right],$$

which can be factored into

$$W = \frac{p_1v_1}{n-1}\left[n - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right],$$

which is a general expression for the work of a stroke with admission and adiabatic expansion, and which can also be transformed to express the work of compressing adiabatically and displacing into a reservoir in air- or gas-compressors.

169. Temperature Changes in Adiabatic Expansion.— Since in adiabatic expansion

$$p_1v_1^n = p_2v_2^n,$$
$$\left(\frac{v_1}{v_2}\right)^n = \left(\frac{p_2}{p_1}\right).$$

Multiplying both sides by $\frac{v_2}{v_1}$, we have

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \frac{p_2v_2}{p_1v_1}.$$

But

$$\frac{p_2v_2}{p_1v} = \frac{RT_2}{RT_1};$$

hence

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \frac{T_2}{T_1} = \left(\frac{1}{r}\right)^{n-1}.$$

But the previous paragraph has shown

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}};$$

hence

$$\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \frac{T_2}{T_1}.$$

Which can be substituted in either of the previous expressions, giving

$$W = \frac{p_1 v_1}{n - 1}\left[n - \frac{T_2}{T_1}\right],$$

a convenient transformation for use in problems connected with compression. While this is of value as presenting general expressions for the relations of temperature and pressure in the pressure-volume discussion, a more direct method is convenient. At the state indicated by p_1 the inherent heat energy will be the product of the specific heat by the absolute temperature. This can be expressed in foot-pounds for the volume v_1 by multiplying the product of cT_1 by the factor $J = 778$, and again by the number of pounds required to fill the volume v_1, which can be called l. Or,

$$E = c \times J \times T_1 \times l.$$

At the end of the expansion, no heat having been given out or received, but all heat energy having been directed to perform external work, and the final temperature is T_2, found from the preceding relations of volume or pressure, it will be true that

$$E_2 = c \times J \times T_2 \times l,$$

whence the work in expansion becomes the difference in the two states of energy, or

$$E_1 - E_2 = W_1 = Jc(T_1 - T_2),$$

or the outer work in expansion is proportional to the difference between the initial and final absolute temperatures, as should have been foreseen.

The above relation also indicates how much heat in units must be withdrawn by artificial refrigeration in compression, or must be added in expansion between limits of v_1 and v_2 or p_1 and p_2 if the final temperature is to be kept the same as the initial temperature, as required in isothermal conditions.

170. Conclusions regarding Isothermal and Adiabatic Expansion.—Neither isothermal nor adiabatic expansion is ever exactly realized in practice, by reason of the conducting, absorbing, and radiating effect of metallic walls of the cylinder in which actual work is done. These interchanges take place so rapidly that differences in piston-speed affect the action less than might be anticipated. True isothermal expansion is not desired where the heat-carrier is a hot medium, because at the end of expansion the gas is as hot as it was at the beginning, and this heat is voided at the exhaust with attendant waste of applied heat. What is desired is to reduce the inherent energy in the medium at exhaust to its lowest practicable value.

In the foregoing discussion, the volume v_1 at the end of admission from the boiler or reservoir of energy has been the quantity assumed, and its pressure p_1 given to it by heat or other source of energy. Both these quantities have been made arbitrary. If it be desired to assume the initial temperature the initial volume v_1 can be taken as that of a unit of weight of the medium, and filling this volume at a calculated pressure p_1 belonging to that state of the medium. The work for l pounds of the medium will be simply l times greater than that for one pound.

171. Design of Cylinders for Piston-motors.—The foregoing paragraphs lead directly to the fundamental processes of the design of a cylinder volume V which will perform a given mechanical work in foot-pounds when the mean effective pressure has been ascertained. In the general formula

$$\text{H.P.} = \frac{PLAN}{33000}$$

the value for P (when usual assumptions are made as to cut-off) is given for the various media by the preceding deductions. The horse-power or foot-pounds is given as a fixed element of the problem, and the desired number of traverses of the piston is imposed as a rule by the work to be done. Hence the remaining factors to be worked out are the relations of L to A when their product LA for one stroke is known. The product LN is called the piston-speed, and is the aggregate of the piston travel in feet per minute. Values for LN as a product will make a high-speed or a low-speed engine according as

$LN = 400$ to 600 ft. per minute......... low speed
$LN = 600$ " 800 " " " .... moderate speed
$LN = 800$ " 1000 " or over, per minute... high speed.

When the value for N indicates a high or a mean or a low speed, the relation of length to diameter of cylinder is likely to be

Stroke = diameter $\times$ ($1\frac{1}{2}$ or 2) for low speed
" = " $\times$ $1\frac{1}{2}$ " mean "
" = " $\times$ 1 " high "

Hence from the relation $A = \dfrac{\pi d^2}{4}$, and the assumption concerning LN, a value for L and for the cylinder diameter can be worked out for a single-cylinder engine. The cylinder proportions having been fixed upon, the design of the valve-gear to give the required cut-off and the proportions of parts to resist the dynamic strains belong to another branch of engineering design apart from the present purpose. The questions concerning clearances with their losses, and the economics possible as affecting wise design, belong to the more advanced treatment of heat-engines in subsequent chapters.

These remarks concerning design apply to the single or simple engine of a typical design. But the engine may be compound or multiple-expansion; it may be rotary instead of reciprocating; the heat medium may be used in a turbine. The mechanical features of these motors have been more fully treated elsewhere than is here possible, but some brief reference is desirable.

172. The Compound or Multiple-expansion Engine.— In the compound engine the steam expands continuously after admission is cut off from the boiler in the high-pressure cylinder. There are three stages of such expansion in the triple-expansion engine, and four stages in the quadruple-expansion. The number of cylinders is not important, but the number of steps or stages is the determining factor.

The diagram of steam effort in the Woolf or tandem-compound engine (Fig. 64) reveals the continuity of the expan-
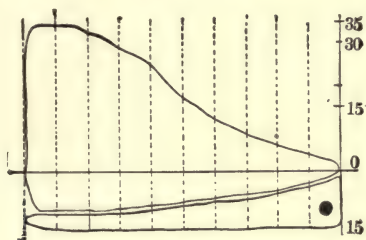
FIG. 64.

sion in the two cylinders, and that the driving steam of the larger or low-pressure cylinder is the back pressure upon the smaller or high-pressure cylinder. The greater area of the low-pressure piston, however, secures a net forward effort. The foregoing formulæ make it plain that it is the final volume of the steam which is significant as compared with its initial volume, so that the cylinder design compels a calculation of the volume of the low-pressure cylinder to secure a required horse-power, and from accepted relations the cylinder area of the smaller is worked out. The other way and more

usual is to draw the P.V. diagram for the desired ratio of expansion, with the given initial and terminal values, and then
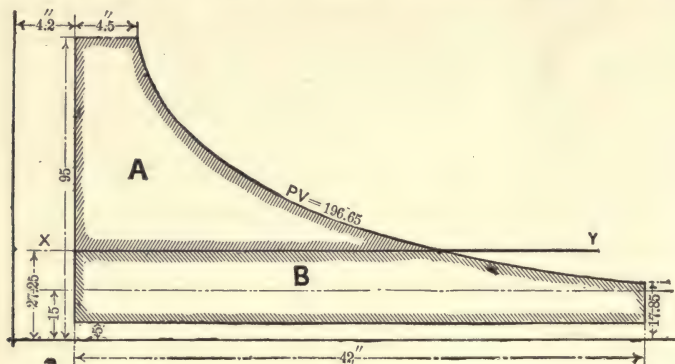
FIG. 65.
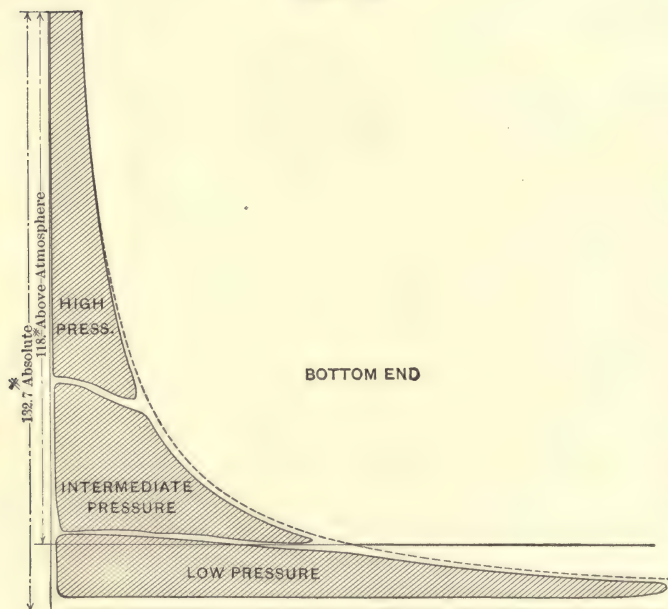
FIG. 66.

divide the area of that diagram, so that equal work will be done in each of the two or more cylinders (Figs. 65 and 66).

In the cross-compound or receiver engine with a cut-off in the low-pressure cylinder there will be an apparent discrepancy or break of continuity in the diagrams of effort. This disappears when the length of the high-pressure diagram is reduced in proportion to that of the low-pressure diagram in the relation of the cylinder-volumes. That is, if the cylinders are as 1 : 4 the length CD of the high-pressure diagram at any pressure level is reduced to $\frac{1}{4}$ of that length at the same level cd (Fig. 67). The only loss of area is from friction or condensation or free expansion into the receiver. It is the
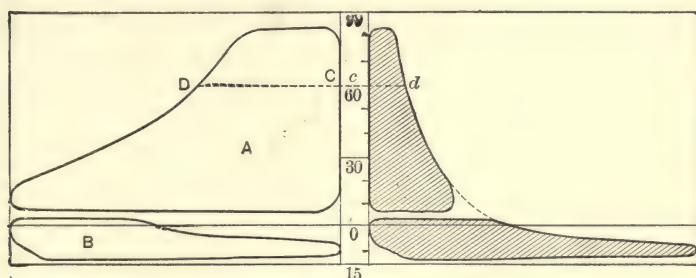
FIG. 67.

loss between the two diagrams which the reheater between cylinders aims to reduce.

The following table presents accepted practice with respect to a selection of the grade of expansion with fixed boiler-pressures:

When the values for T are those which belong to a
    pressure below 80 lbs.............. use single engine
    for pressures between 80 and 100 lbs. " compound engine
    "   "   "   130 " 160 "   " triple   "
    "   "   above 170 lbs........ " quadruple "

Usual cylinder-ratios of practice, for usual pressures with triple engines, are:

Pressures.	Small.	Intermediate.	Large.
130............	I	2.25	5
140............	I	2.40	5.85
150............	I	2.55	6.90
160............	I	2.70	7.25
170............	Quadruple engine preferred.		

For quadruple-expansion engines the usual ratios of cylinder-areas and volumes approximate 1 : 2 : 3.78 : 7.70, which may be called 1 : 2 : 4 : 8.

If the principle be adopted that the ratios of areas are to be as the fourth root of the number of expansions, the ratio of the first to the fourth will be as the cube of the fourth root. The ratio will increase as the initial pressure becomes greater; e.g., 1 : 2.2 : 4.8 : 10.6.

Mr. G. I. Rockwood has designed a compound engine with a cylinder-ratio of 7 : 1 with the view of making heat-range equal in the two cylinders, whence the ratio of surfaces is taken account of, as well as the differences in temperature.

The subjects of the mechanisms of the compound engine, arrangement of cylinders, compounding above atmosphere, the compound locomotive, etc., will be found discussed elsewhere, to which references appear in the Appendix.

173. Advantages of the Compound or Multiple-expansion Engine.—The principle of securing expansion by the continuous working of steam in cylinders of increasing volume is to be defended by reason of the following advantages:

1. The high grade of expansion and the difference between the initial and final temperature in the steam used is secured with an admission of steam into the cylinder through a longer proportion of the stroke than in the single cylinder. It has been seen that the efficiency of the fluid used increases with the difference in the initial and final temperatures. The work of the steam reaches the crank in angles more favorable to produce rotation.

2. With the terminal temperature at exhaust fixed by the temperature possible with the means used to condense the steam, the compound principle enables higher pressures to be used in the boilers as initial pressures in the cylinder. To increase the pressures in the boilers is to carry more stored energy in a given space; to use higher pressures is to enable each cubic foot or pound of steam to carry more energy into the engine-cylinder, and the given quantity of heat raises the pressure of steam more rapidly after the steam has become a complete gas than it does at lower pressures, when a large part of the heat is absorbed in changing the molecular condition of the water.

3. By receiving the high-pressure steam from the boiler first upon a cylinder of small area, as in the compound engine, the strain upon the mechanism at the joints and moving members is less than if that same pressure had to be received at the beginning of a stroke in a cylinder, and against a piston of a large diameter. Less loss from friction also follows during the less effective angles of the stroke.

4. From the longer period of admission discussed in number one above, it follows that a more advantageous arrangement for admitting and cutting off the steam becomes possible. With the single cylinder and early cut-off in it, the openings to admit steam would have to be closed so early that it would be difficult to admit steam through wide and generous ports or passages. Such single-cylinder valve-gear with narrow areas for steam would introduce the difficulty known as wire-drawing of the steam. This is a phenomenon present when the pressure of steam is reduced by compelling it to pass through a narrow or constricted opening.

5. With high-pressure steam it is difficult, both by reason of changes of shape due to heat and by reason of the pressure itself, to make the valves controlling the admission of steam so that they shall be and remain tight. In the compound engine the steam which leaks past the valve of the first

or high-pressure cylinder does not leak into the air or condenser, but into a later cylinder in the chain in which it expands and does work.

6. If by reason of doing work in expanding there is a transformation of heat into work which must be compensated by a condensation of the steam in the first cylinder, that water reheated and expanding at the lower temperature does work in the later cylinder of the chain instead of escaping unutilized through the exhaust.

7. In those forms of the compound engine in which the work of the several cylinders reaches the crank-shaft each through its own crank-pin, there is the advantage of such distribution, for this avoids the concentration for large engines of great energy on small areas, and enables designers to avoid either excessive lengths or inconvenient diameters for their crank-pins. When the crank-pin becomes of inconvenient diameter with respect to the length of the crank, the angle during which the pressure of steam is available to produce rotation of the crank is diminished.

8. The turning effort is equalized when the compound engine is arranged to have its cranks quartering. This diminishes the weight of the fly-wheel.

9. The compound engine gives an opportunity to improve the quality of the steam during the process of expansion when it is possible to use a reheater.

10. The clearance-volumes of the small-diameter cylinder carry less steam by weight than if the steam had to fill the clearance-volume of the large cylinder. The steam in these clearance-volumes is also used expansively in the later cylinder, instead of being rejected, as would be the case in the single cylinder.

11. The hottest steam is used in the cylinder of the smallest volume, causing a diminished loss from radiation and condensation due to cool external air.

12. In the compound locomotive the less terminal pres-

sure gives less violence to the escape of the exhaust which induces the draft through the fire-box. The fire is pulled about less.

13. The greatest advantage incident to the use of the principle of continuous expansion in several cylinders is that thereby the range of temperature between the initial and final states of that cylinder is less than it would have to be if the expansion were in the one cylinder only. The law of transfer of heat from one body to another is that the transfer is rapid in proportion as the difference in temperature is greater. The less the temperature between the incoming and outgoing steam in any cylinder, the less condensation occurs when the hot steam enters. This is a particularly favorable condition for the large and low-pressure cylinder, whose ends are alternately open to the comparatively low temperature of steam as it is escaping into the condenser. It is of great advantage that the high-temperature steam fresh from the boiler should not have to meet the relatively cool metal and large surface of this low-pressure cylinder.

174. Disadvantages of the Compound Engine.—When it is recalled that the low-pressure cylinder is the fundamental unit, and determines the working capacity of the compound engine, it is apparent that by introducing the other cylinders in the multiple-expansion type certain disadvantages are introduced. These are:

1. The cost of the cylinders other than the low. This may mean in tandem engines the cost of piston and cylinder with additional rod, but in cross-compound and fore-and-aft engines it means an additional cost of practically another engine with crank, connecting-rod, cross-head, and the like.

2. The weight and bulk of the additional cylinder adding to foundations and taking up valuable space.

3. The friction-loss due to the work absorbed by this extra cylinder in operating its mechanism, valve, and the like.

4. The loss by radiation of heat from the surface of the extra cylinder and valve-chest, which are surfaces exposed to the air.

5. The loss of work due to the difficulties represented by lost area in the work-diagram from friction, free expansion, condensation, and the like. The single-engine diagram, getting the same grade of expansion in the same cylinder, would not experience this.

6. The difficulty connected with regulating the power of the engine when the work varies widely, and the first cylinder has measured off a volume of steam adapted to a resistance different from that upon the engine when that volume of steam reaches the later cylinders. This is the difficulty of regulating the multiple-expansion engine, except by regulating devices operating upon each cylinder independently.

7. There has been considerable trouble in compound engines from the accumulation of water in the low-pressure cylinders, particularly when compounding above the atmosphere and using wet steam. The wide range of expansion, the lowered terminal pressure, and the large diameter of the low-pressure cylinder have made this difficulty a very troublesome one in locomotive practice.

8. In compound locomotives, the terminal pressure may not be high enough to give intensity to the draft in the stack sufficient to keep the engine steaming freely.

It is very obvious that the weight to be attached to the above objections is not considered by most designers to be great enough to overbalance the advantages which follow from the principle of compounding.

175. Design of the Rotary Engine.—In the rotary engine, a series of pistons or vanes are attached radially upon crank-arms so as to receive the pressure of the steam directly to produce rotation. It is so difficult to secure expansive working by allowing the steam admitted to the cylinder to lower its pressure while doing work, that it is not usually

attempted, and the foregoing formulæ are not applicable. The pressure from the boiler acts upon the effective area of the vane, and moves it through a space in feet (Fig. 68). The product is the foot-pounds exerted while that vane was in action. For the mechanisms and for the advantages and disadvantages of the rotary engine, see the reference
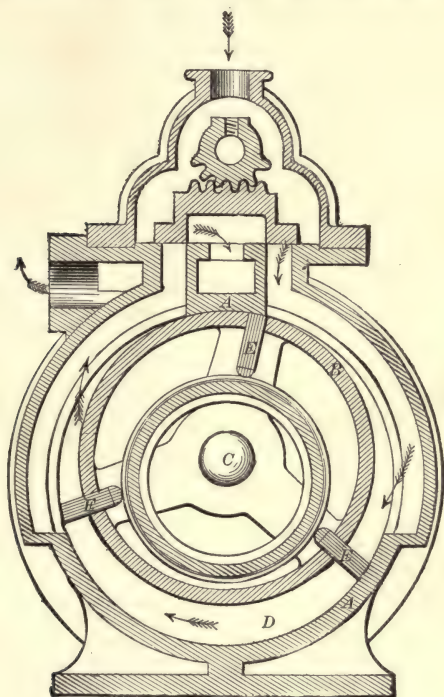
FIG. 68.

in the Appendix. A difference between the initial and final volumes of the steam in the machine itself is only to be easily secured by compounding, or the use of engines in series.

176. The Steam Turbine.—The third type of motor which is used to avail of the pressure energy of an elastic heat medium is the steam turbine. It is a transfer to the field of the elastic fluids of the idea long familiar in water-

motors which use an incompressible medium. The turbine
uses the impact of a weight of fluid having a velocity due to
head or equivalent pressure, and after the impulse has been
utilized, a further propelling effect is sought from the reac-
tion of the current of the fluid.

The low density of steam as compared with water makes
it difficult to secure a high value for the unit of effort upon
the bucket of the steam turbine; the best results along this
line are secured by using a considerable number of jets of
steam. On the other hand, if it be conceded (as can be
proved) that the best result is obtained when the velocity of
the jet is twice that of the surface receiving its impulse, or
when the circumference of the wheel is moving tangentially
with half the velocity of the steam at the nozzle, the wheel
has to have so high a velocity that it is not easy to resist the
centrifugal effort of the outer elements of the wheel itself.
For example, if the steam issue from the nozzle at a pressure
of 140 pounds, it has a velocity of upwards of 2000 feet per
second. This follows because $v = \sqrt{2gh}$ and $h = (140 \times 144) \times 0.3148$, the latter being the weight of a cubic foot of
steam at that pressure. Solving when $h = 64{,}000$ feet, v
becomes 2030 feet per second.

This linear velocity at the circumference compels a rota-
tive speed of many thousand revolutions per minute for the
turbine wheel proper, which has to be of small diameter to
withstand the centrifugal strain, and therefore a reducing train
to bring the speed to limits which are convenient and usual.
On the other hand, where high rotative speed is no disadvan-
tage, as in driving of dynamo-armatures or specially designed
propeller-wheels, the turbine is the most compact and light-
est of motors.

The mechanical principle underlying the motors of this
class is that the energy imparted to the wheel will be the
difference between the moments of the living force of the
mass of the fluid upon entering and leaving the wheel, mul-

tiplied by the angular velocity of the wheel. That is, if x be the tangential velocity of the jet, and u be the less tangential velocity of the buckets, the work per second will be

$$\text{Work per second} = M(x - u)u,$$

when $M = \dfrac{W}{g}$ is the mass passing the nozzles in one second, and $u = \omega r$. If the steam left the wheel without velocity relatively to the direction of its effort, the work per second per pound would be:

$$\text{Work per second per pound} = \frac{u^2}{g},$$

when the velocity of the rim is known. If the weight in pounds of steam is known which flows per second (w) and the work per second per pound is

$$U = \frac{1}{g}(x - u)u,$$

then the horse-power per minute becomes

$$\text{H.P.} = \frac{wU \times 60}{33000}.$$

The outward-flow turbine (Dow type) permits the range of pressure to occur within the arm, so that at the issue-point the steam has expanded to atmospheric pressure or nearly so, leaving only energy and velocity enough to free itself from the wheel. The same is secured in Parsons and Curtis turbines by the principle of compounding, whereby successive increase in volume results in the lowering of pressure. In the De Laval turbine the expansion occurs in the nozzle itself by the changes in its cross-section.

Figs. 69a, 69b, and 70 show typical sections of steam-turbines selected from outward-flow and tangential impulse

beginning with the
inner series every
alternate one is a
turbine; the others
are guide plates.

Fig. 2.

FIG. 69*a*.

Tachometer
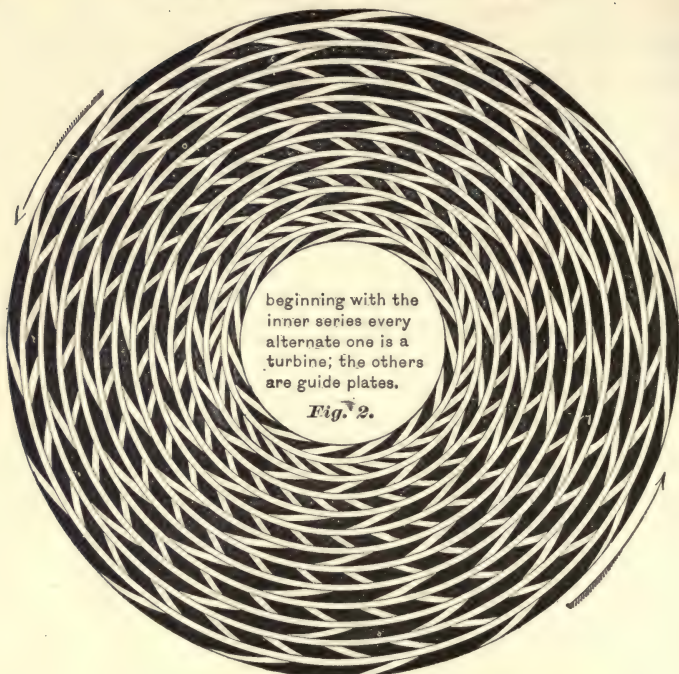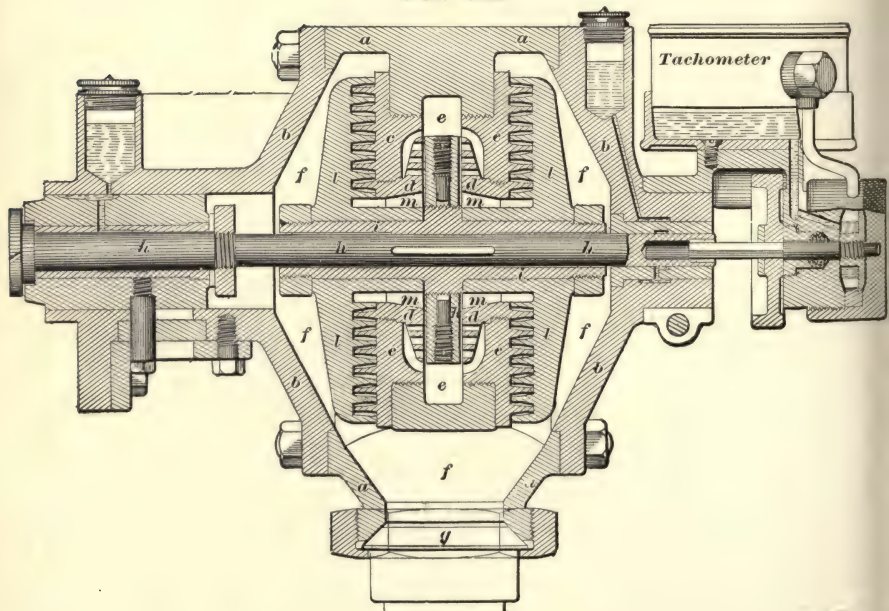
FIG. 69*b*.

types. The high speeds compel the greatest care in balancing and in the construction of the bucket detail. Where it can be done, the plan of mounting the axis of the turbine verti-
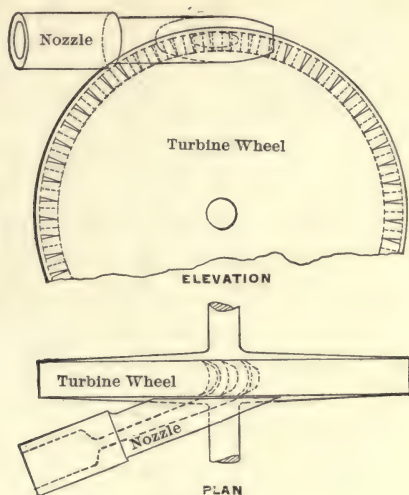
FIG. 70.

cally avoids the disturbance of balance from its own weight. Excellent results have been secured by the use of an annular jet around a fine central needle in a small nozzle.

CHAPTER XIII.

MECHANICAL COMPRESSION OF HEAT MEDIA.

180. Introductory.—It will be at once apparent that in the case of permanent gases or heat media within the usual limits of pressures, it is easily possible to invert the formulæ for expansion and so express the changes when a heat medium has outer work exerted upon it to produce those changes in its heat condition which must follow its subjection to such external force. When such changes are studied upon the pressure-volume plane the diagram of pressures is traced in reverse direction, or contrary to that of the hands of the clock in the previous right-hand diagrams. When this is done with air or gas as a medium, the machine is called an air- or a gas-compressor. The compressor is a heat force-pump, raising a weight of air to a higher heat condition by converting external work into heat. The blowing engine differs from the compressor only by having a low ratio between final and initial pressures of the medium, and usually also by being adapted to handle large volumes of air.

181. The Air-compressor, with Pressures Given.—The air-compressor problem in practice usually requires that air shall be taken into the cylinder at a pressure (p_1) of the atmosphere or nearly, and that a volume per stroke (v_1) of air at this pressure shall be compressed to a higher pressure (p_2). This will be accomplished by reducing its volume to (v_2) corresponding to that higher pressure, when the valves leading to a receiver will be opened by the pressure in the cylinder,

and the compressed air without increase of pressure or tem-
perature will be forced out into the receiver. If the air un-
dergoing compression is cooled by water-jacketing or by injec-
tion of water, the compression may be isothermal. As a
rule, however, by reason of the short time allowed for the
compression and the slowness of transfer in air, the compres-
sion is practically adiabatic, and the air is warm or hot as it
leaves the compressor. If the air is only compressed to re-
ceiver pressure, at the end of the piston traverse, the valves
will not open to discharge the air, and consequently on the
return traverse the compressed air in clearances will simply
expand down to atmospheric pressure (p_2) like a spring and
no fresh charge will be taken in. It will be apparent, there-
fore, that clearance volumes are of significant detriment in air
compressing, and the higher the pressure above atmosphere,
the worse the loss which they occasion.

The formulæ of § 167 are therefore directly applicable,
provided the direction of the process be reversed, in the dia-
gram representing the cycle. The total work of one stroke
will be made up of:

$$\text{Work} = \left\{ \begin{array}{c} \text{compression} \\ \text{work} \end{array} \right\} + \left\{ \begin{array}{c} \text{displacement} \\ \text{work} \end{array} \right\} - \left\{ \begin{array}{c} \text{inlet back-} \\ \text{pressure} \end{array} \right\}$$

$$W = \qquad W_c \qquad + \qquad W_d \qquad - \qquad W_i$$

$$W_c = \int_{v_2}^{v_1} p\,dv$$

$$= p_1 v_1 \int_{v_2}^{v_1} \frac{dv}{v^n} = \frac{p_1 v_1 - p_2 v_2}{n-1}$$

$$= \frac{p_1 v_1}{n-1}\left[\left(\frac{v_1}{v_2}\right)^{n-1} - 1\right],$$

which can be transformed as in § 168 to

$$W_c = \frac{p_1 v_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

The work of displacement

$$W = p_2 v_2 = p_1 v_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}},$$

by using the same methods as in the foregoing expansion discussion, since

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n;$$

by multiplying both members by $\frac{v_2}{v_1}$ we have

$$\frac{p_2 v_2}{p_1 v_1} = \left(\frac{v_1}{v_2}\right)^{n-1},$$

whence

$$p_2 v_2 = p_1 v_1 \left(\frac{v_1}{v_2}\right)^{n-1} = p_1 v_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}.$$

The compressing cycle work is therefore

$$W = \frac{p_1 v_1}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + p_1 v_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - p_1 v_1,$$

which factors into

$$W = \frac{p_1 v_1 n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

This has been made intentionally to differ from the form of the statement for expansion work (§ 167), inasmuch as in expansion the back-pressure line does not necessarily coincide with the end of the expansion line, and to introduce another pressure value not related to the rest of the factors would be to complicate the computations. In compression, however, the initial pressure is the same as the terminal pressure for the intake stroke.

182. The Air-compressor with Volumes given.—If the volume of compressed air should be given, or the volume at atmospheric pressure, the relations between pressures, volumes, and temperatures given in §§ 167 and 168 enable easy substitutions to be made. In adiabatic changes

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\frac{1}{r}\right)^{n-1} = \frac{T_2}{T_1},$$

in which r expresses the relation between the less and greater pressures, and the greater and the less volume. The relation shows how the temperatures absolute will change as external work is done upon the air, and also how much heat must be abstracted if the compression is to be kept isothermal and the air to be at the same temperature in the receiver as in the outer air. The table on page 264 gives data of interest as to the rise in temperature for air.

183. Value of the Factor n in Air-compressing.—The exponent n of an adiabatic expansion and compression of air or other non-condensable gases will be the ratio between the specific heat of the gas at constant volume and the specific heat at constant pressure; i.e.:

$$n = \frac{C_p}{C_n}.$$

For air, when $C_p = .238$ and $C_v = .168$

$$n = \frac{238}{168} = 1.41.$$

The heat required to raise a unit weight one degree at constant pressure will be obviously the greater since the expanding air is overcoming the outer pressure on it as it increases in energy but does not increase in pressure. The difference $C_p - C_v$ will denote the amount of heat corresponding to overcoming the pressure p_0 when a volume v_0 of air at zero degrees is heated one degree.

One hundred volumes of dry air at mean atmospheric pressure of 14.7
pounds per square inch and a Fahrenheit temperature of 60° (15.5 C.) when
compressed without withdrawal of heat will have the temperatures centi-
grade given in column 2 of the following table, and the volumes given in
column 3 for that pressure. If the compression is isothermal so that the
temperature is kept at 60° F. or 15.5 C., the volumes will be as in column 5.
(Thurston, *Journ. Frank. Inst.*, 1884.)

Pressure Absolute, Pounds per Square Inch.	Temperature Centigrade at end of Compression.	Volume of Temperature and Pressure Preceding.	Temperature Fahrenheit Corresponding.	Volume if Temperature had remained constant at 15.5° C.
1	2	3	4	5
14.7	15.5	100.0	60	. . . .
15	17.26	98.58	63	98.00
20	42.60	80.36	108	73.50
25	64.76	68.59	149	58.80
30	82.10	60.27	180	49.00
35	98.38	54.01	208	42.00
40	113.86	49.13	237	36.75
45	126.54	45.18	259	32.67
50	138.96	41.93	282	29.40
55	150.53	39.19	303	26.73
60	161.38	36.84	322	24.50
65	171.61	34.80	340	22.62
70	181.29	33.02	357	21.00
75	190.49	31.44	375	19.60
80	199.26	30.03	391	18.38
85	207.66	28.77	405	17.29
90	214.71	27.62	418	16.33
95	223.25	26.58	434	15.47
100	230.91	25.63	447	14.70
125	264.66	21.88	508	11.76
150	293.91	19.22	561	9.80
175	319.87	17.23	608	8.40
200	343.31	15.67	649	7.35
225	364.71	14.41	687	6.53
250	411.57	13.38	772	5.88
300	420.34	11.75	788	4.90
400	480.76	9.58	896	3.90
500	531.21	8.17	986	2.94
600	574.93	7.18	1065	2.45
700	603.74	6.44	1117	2.10
800	648.80	5.86	1200	1.84
900	680.86	5.39	1256	1.63
1000	710.49	5.00	1310	1.47
2000	929.67	3.06	1706	0.74

But it has been already seen (§§ 107 and 115) that

$$\frac{p_0 v_0}{T_0} = \frac{p_1 v_1}{T_1},$$

or that

$$p_1 v_1 = RT_1.$$

Hence it follows (see § 116) that

$$J(C_p - C_v) = R.$$

From which, if the other quantities have been observed, the value of the mechanical equivalent can be calculated in advance of experiment. When densities are known, for any standard temperature, R can be calculated for any gas, since

$$R = \frac{p_0 v_0}{T_0} = \frac{p_0}{D_0 T_0},$$

because the weight of a unit cubic volume is by definition its specific gravity. (See § 107).

This enables a usual transformation of the equation for work to be made. For since

$$C_p - C_v = C_v(n - 1),$$

$$n - 1 = \frac{R}{JC_v};$$

hence (§ 167)

$$W = \frac{Jp_1 v_1 C_v}{R}\left(1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right.$$

when there is no work of admission.

184. Mean Pressure in the Compressing-cylinder.—If the work of the compressing-cylinder be given by a $p_1 v_1$ product, which can be represented by an area, the mean pressure will be the height of a rectangle whose base is the final vol-

ume. Hence the mean pressure will result if the equation for W be divided by v_1. This makes (§ 180)

$$\text{Mean effective pressure} = \frac{p_1 n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

If the compressor is a steam-machine, the mean effective pressure in the steam-cylinder must be enough greater than this to overcome the frictional resistances at bearings, guides, stuffing-boxes, etc., and to overcome the friction in the cylinders from packing-rings, valves, and the like, and the fly-wheel must be able to equalize effort and resistance by its stored energy when these are unequal at different parts of the stroke.

185. Isothermal Compression.—The formula for isothermal expansion (§ 166) and for mean pressure for such expansion are the same for compression, with proper changes in the letters for volumes and pressures. The formula becomes

$$W = p_2 v_2 + \int p\,dv = p_1 v_1 + p_1 v_1 \text{ Nap. log } \frac{v_1}{v_2}$$

$$= p_1 v_1\left[1 + \text{Nap. log } \frac{v_1}{v_2}\right],$$

and the mean pressure as before is found by dividing by the length or volume v_1; or

$$\text{M.E.P.} = p_1\left[1 + \text{Nap. log } \frac{v_1}{v_2}\right].$$

When this is compared with the work and M.E.P. in adiabatic compression, it will be found that isothermal compression requires less work than adiabatic compression between the same pressures, but a less volume is displaced into the receiver because of the reduction of volume resulting from the abstraction of the heat of the compression, which is made to disappear by the cooling and is lost. That is, if Fig. 75 represents by its curve of compression *ia* the path of the

pressure-volume relation in adiabatic compression, and by its curve *is* the path of isothermal compression, there is a waste of power in adiabatic compression represented by the shaded area *isa* as compared with isothermal, when both compressions start from atmospheric pressure and temperature at *c*. If the air cools down from the higher temperature belonging to the adiabatic point *a* to the atmospheric pressure in pipes and reservoirs so as to have a volume represented by *es* instead
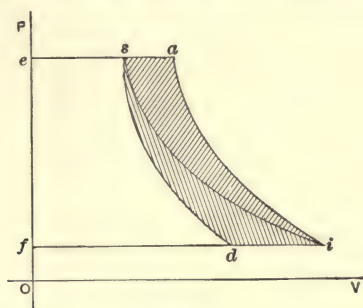
Fig. 75.

of *ea*, the adiabatic compression will suffer a further loss when the air comes to expand from the volume *es* back to atmospheric pressure in a proper air-engine (Chapter XVIII). The expansion, if adiabatic and starting from *s*, will end at a point *d*, while, if isothermal, would return to *i*. Hence the area between the bounding curves *sd* and *ai* will represent the waste of power if the compression were adiabatic as well as the expansion. If the compression were isothermal, the loss would be only the area *isd*. If the expansion could be also isothermal, there would be no loss in the reversible process outside of pipe friction (see further, Chapter XVIII).

186. Effect of Clearance in Compressing-cylinders.—It has been already said (§ 181) that any air remaining behind in a compressing-cylinder, between piston-head and valves, will expand on the return or inlet stroke, preventing early opening of the inlet valves or any flow of air into the cylinder, until

the pressure of that imprisoned air falls below that outside
which acts on the inlet valves to open them.   The effect of
this expanding air is to help the working stroke at the start,
by acting as a cushion or spring, but the harmful effect is to
increase the size of cylinder required for a certain net output
of air.   The expansion of this imprisoned air is safely called
adiabatic, and the work which it does (if it is worth while to
find it) is to be found for the weight or volume as given in
the expansion value for W (§§ 167 and 168).   The increase
in the size of cylinder required is to be found by plotting the
curve of such expansion work on a diagram (Fig. 76), and in-
creasing the volume by that percentage of itself which is given
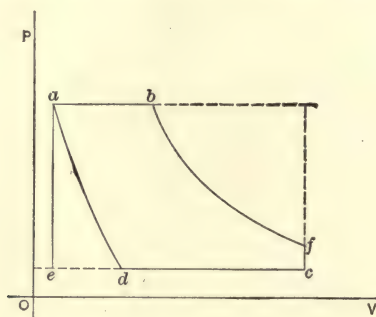
FIG. 76.

by the ratio of the entire length of the diagram between ver-
ticals *ec* to the length *cd* of the diagram between the termi-
nals of the adiabatics.

The mean effective pressure is of course reduced by the
effect of the clearance volume of air.   This will be in the pro-
portion of the effective length between the feet of the adia-
batics to the total length between perpendiculars.   Other-
wise, the formula for mean effective pressure may be applied
directly in calculating the work to be done in horse-power if
the actual volume of air be used, taken after the inlet-valves
have opened, instead of the full piston-displacement.

187. Volume of the Compressing-cylinder.—If the problem be given to design a cylinder to give V_2 cubic feet of air at a pressure of p_2 pounds per square inch (or per square foot) after compression, it will first be necessary to find the corresponding absolute temperature T_2 corresponding to that pressure, from the foregoing equations. Then the volume V_1 at atmospheric pressure and temperature can be found from the relation

$$\frac{p_2 V_2}{T_2} = \frac{p_1 V_1}{T_1}.$$

Then if clearance effect be neglected, and the piston makes $2n$ traverses per minute for n revolutions in that same time, the cylinder volume v_1 will be

$$v_1 = \frac{V_1}{2n} \text{ cubic feet.}$$

If there be a clearance expressed in terms of piston-displacement by the fraction $\dfrac{1}{c}$, the air in the clearance volume expanding down from p_2 to p_1 will occupy a volume which will be

$$\frac{1}{c}\left(\frac{p}{p_1}\right)^{\frac{1}{n}}$$

of that piston-displacement. The denominator $2n$ should be therefore multiplied by the factor

$$1 + \frac{1}{c} - \frac{1}{c}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

of itself to allow for this clearance loss. Furthermore, the inlet-valves do not open until the pressure in the cylinder is less than p_1 and friction through them keeps it less than p_1 on the aspirating stroke; and similarly, the delivery-valves

do not open until p_2 is exceeded, and their friction makes the pressure of displacement in the cylinder greater than p_2. These losses compel the real volume to exceed the calculated volume by an amount for which experience is the only guide.

188. Cooling of Compressing-cylinder.—The relation between pressures and absolute temperatures in compressing

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

enables the rise or range of temperature to be calculated when the pressures are given. Hence if it be desired to cool the compressing-cylinder by water-jacketing or to cool the air by injection of water, the pounds of water required are easily calculated. If t_a is the range of temperature for the air, and t_w is the range to be permitted to the water, then the law of transfer gives

$$1 \times C_v \times t_a = w \times 1 \times t_w$$

for each pound of air, when w is the desired weight of water with a specific heat of unity.

Water-jacketing does not cool the air much, since air is cooled by contact only, and this is not easily managed unless the air is in thin films. Injection of water is effective for cooling the air, but in many cases the presence of water-mist in the compressed air is an objection on account of its freezing when the air is used expansively in the compressed-air motor. If the air be assumed to be saturated with all the steam-vapor it can carry, the effect of the steam-vapor is probably inappreciable on the work of the air. The exponent of the equation may be slightly affected, but this is all.

189. Compressing in Two or More Stages. Compound Compressors.—It early suggested itself to designers of compressors that if the compound principle were applied to compressing they would reap certain advantages belonging to the principle as applied to the steam-engine. The air taken

at lower pressure and larger volume into a large cylinder and
delivered from it to a smaller one would allow an inter-
cooler between the two cylinders to be advantageously ar-
ranged to cool the air and diminish the volume of the small
cylinder, while the entire volume of the small cylinder was
available for the second stage of the compression, instead of
forcing this into that part of the stroke of the one cylinder at
which the diminishing crank-angle reduced the piston to a
slower velocity. The intercooler, however, has a manifest
advantage, because it usually happens that a storage of air in
reservoirs occurs in which the temperature of the air drops
down to or near that of the incoming atmospheric air. Here
the work of compression can be reduced by the division into
two stages, by making it approach to isothermal compres-
sion. The air is taken in at p_1 and is compressed in the first
cylinder to p_2', and in the reservoir at p_2' it is cooled back to
T_1 or nearly so, at which it entered the first cylinder. The
second cylinder draws the air at a pressure p_2' and com-
presses it to the final p_2. This is represented graphically for
a three-stage compression by Fig. 77. By the first cooling
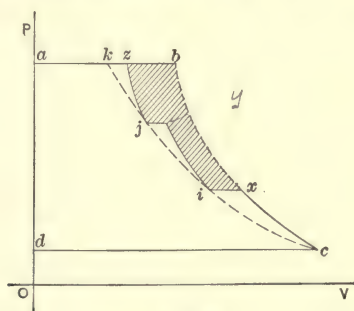
FIG. 77.

the volume is diminished from the adiabatic point x to the
isothermal point i. The second compression adiabatic in the
second cylinder would bring the pressure and volume to y;
by the second intercooler it is reduced in temperature, and

The hatched
area is the saving
made by 3 stage compression over
one stage compression etc. Durham.

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therefore in volume, to j. The final compression carries the pressure up to the volume-line through z, whereas k would have been the isothermal point. Hence the loss drops to that represented by the hatched area instead of being the full area outside of the isothermal curve $cijk$.

What is desired in two-stage compression is to make such a division of the compressing work as shall make it a minimum. If the first stage compress from p_1 to p_2' and the work be called W_1, the equation of § 181 is

$$W_1 = p_1 v_1 \frac{n}{n-1}\left[\left(\frac{p_2'}{p_1}\right)^{\frac{n-1}{n}} - 1\right],$$

for one pound of air, neglecting clearance loss.

The work of compressing from p_2' to p_2 is

$$W_2 = p_2' v_2' \frac{n}{n-1}\left[\left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}} - 1\right].$$

But if the air be cooled back to the temperature T_1 in the intercooler between the cylinders, this last work becomes

$$W_2' = p_1 v_1 \frac{n}{n-1}\left[\left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}} - 1\right].$$

Hence the total work in the two cylinders is their sum; or,

$$W = W_1 + W_2' = p_1 v_1 \frac{n}{n-1}\left[\left(\frac{p_2'}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}} - 2\right].$$

This value for W will be a minimum when

$$\left(\frac{p_2'}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}}$$

is a minimum. When p_1 and p_2 are known, this expression is of the form $\dfrac{x}{a} + \dfrac{b}{x}$, which can be differentiated, and when

the first differential coefficient is put equal to zero, the value for x corresponding to such minimum $= \sqrt{ab}$. Hence the minimum value for the work occurs when

$$p_2' = \sqrt{p_1 p_2}.$$

When the air is supplied to both cylinders at the temperature T_1 their respective volumes should be inversely as the absolute values of the pressures p_1 and p_2'.

When high pressure is sought, a similar reasoning would point to using three stages or four. The smaller diameter of the higher pressure cylinders enables greater strength to be secured with less proportionate increase in weight, besides the diminution of the motor work, resulting from cooling the air in transit. The intercooler is usually a receiver with a coil of pipe within it, around which the air passes.

190. Fluid Compressors.—By combining the use of a displacing-piston with the use of a displacing fluid in the compressing-cylinder, or by using a fluid alone on which the air or gas undergoing compression was without physical or chemical effect, designers have been able to diminish the evil effects of clearance. The liquid used may be water or oil. It fills all dead space behind the piston up to the valves, which are placed above the liquid, and therefore the expulsion or displacement of air is practically complete. The pumps must not operate at too high a speed with these fluid pistons, which must not be allowed to churn or spatter. If water is used, it grows warm and begins to form vapor, and requires renewal.

191. Conclusions and Remarks.—It will have been made apparent that in compressing air, as the volumes diminish, the pressures increase more rapidly. Air at very high tension, therefore, is not heated so hot in proportion to the stored energy in it as at the lower pressures. It is the loss of the heat of compression in receivers, pipes, conduits, and the like which withdraws energy from compressed air after it is
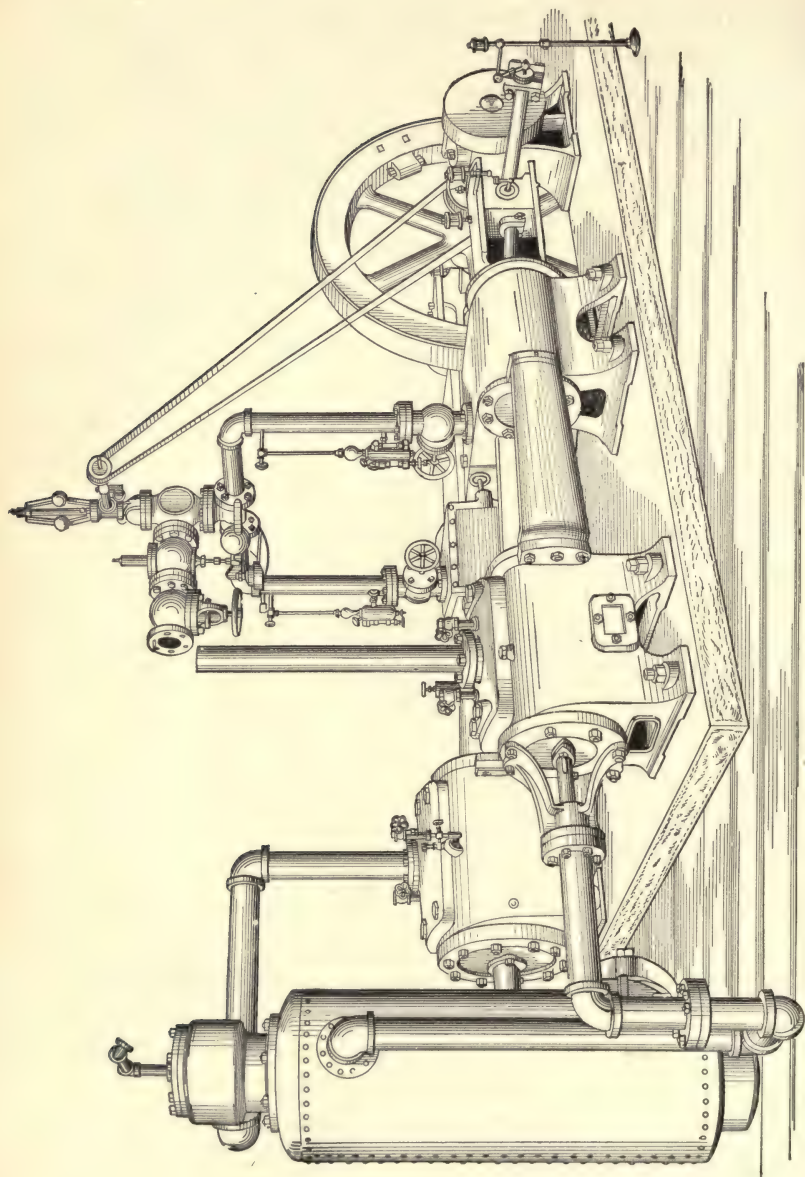
FIG. 78.

stored in it by the compressing process, which forces compressed air to lower effectiveness as a medium of transmitting energy, unless the air is heated again just before using, which can be cheaply done. The cooler the intaken air, the greater the weight per cubic foot and the greater the mass or weight of air handled by a machine of a given size. Further, the greater energy is a given machine capable of imparting to the air. At high mountain altitudes the efficiency of a compressor is less than at sea-level. If the air can be used directly as compressed, the heat energy stored in it is of more moment than the pressure without the heat. If the air is not used expansively in the air-motor, energy stored in it is wasted at the exhaust. If the air is used expansively, its condition as to heat is lowered, and the energy expended in doing useful work. If pressure is raised without heat-rise, then in working expansively its heat standard is lowered below the normal, and it has to be regenerated.

The operation of the air-engine, or compressed-air motor, and the storage losses, will be discussed in Chapter XVIII. Fig. 78 will serve as a type for two-stage tandem steam air-compressors.

CHAPTER XIV.

TEMPERATURE-ENTROPY DIAGRAMS FOR HEAT-ENGINES.

195. Introductory.—It will have been observed that the formulæ and diagrams of the foregoing chapters have been diagrams of work in terms of co-related pressures and volumes, and similar to the diagram traced by the pencil of the steam-engine indicator.

It has long been a conception of the master thinkers on the mutual relations of pressure, volume, and temperature for any medium, that these factors might be regarded as the co-ordinates, taken with respect to three rectangular co-ordinate axes, of points upon a surface, which they have called the *thermodynamic surface* for any medium undergoing such changes of pressure, volume, and temperature as were represented analytically by the equations of the mathematical treatment. If the axis of x be the line representing volumes, and the axis of y represent pressures, the axis of z will be the axis of temperatures. The pv diagram is therefore drawn on a surface parallel to the plane fixed by the axes of p and v, either with disregard of temperatures or with the temperature assumed constant. The distance from the plane through the axes of p and v is the value of the temperature at the pressure assumed for the initial or final pressure. In all the foregoing discussion the variations of temperature have been deduced analytically from the pv variations at assumed points. It would be an interesting deduction if it should result that the exponent of an expansion curve should prove to be the consequence of the distortion of the pv sur-

face when temperature alters with variations of pressure and volume. The projection of pv surfaces will be by lines parallel to the temperature axis.

The projection parallel to the volume axis on the plane through the t and v axes of the pv diagram when the latter is distorted as required by the temperature change in adiabatic expansion or compression gives lines or areas on the pt surface, which, however, are of no practical service. The actual cycle of a heat-motor always demands the addition to the working substance or heat medium of a quantity of heat energy which may or may not take the form of increased temperature. The pvt thermodynamic surface is adapted primarily for the study of phenomena involving no change of energy from without, but only transformations in which the unit weight has the same intrinsic total energy but undergoes only variation in the factors. What is desired is a scheme of graphical representation, whereby the expense of heat in the form of temperature or other form of heat energy can be represented by an area, the product of two factors, which shall be so connected to the pv diagram that the heat expenditure or return which accompanies the cycle of a piston-motor can be readily examined, or the heat work of different motors and media examined and compared, even as the pressure work is studied for design when a capacity in foot-pounds is desired. In gas-engines, or such as have the heat-energy liberated by combustion in the motor-cylinder directly, the temperature phenomena are more significant than the phenomena of the indicator-card in studying efficiencies. While it is also true that certain areas of work expressed in foot-pounds by the pv diagram can be translated into heat-units by the division by $J = 778$, the foot-pounds corresponding to one heat-unit, this can only be done with isothermal expansion of the permanent or true gases and is not exact for isothermal work of vapors or mixtures.

196. **The Temperature-entropy Diagram.**—It will be

recalled, however, from the discussion of thermal lines and the significance of the entropy, that when a heat medium is undergoing the operations usual to such media under the isothermal conditions, with pv a constant product, and the temperature T is not allowed to vary when the pressure or volume vary inversely together, there is a quantity which should vary when increased heat energy is imparted. If this quantity be called the entropy (§ 124), and a quantity of heat in units be added which would raise the absolute temperature through $T_2 - T_1$ degrees, the total increase in entropy for the quantity of heat energy added will be expressed by

$$\int_{T_2}^{T_1} \frac{\delta Q}{T}.$$

Hence when a quantity Q of heat-units is being added to the medium at a temperature T, the energy which is being added can be expressed by the product

Heat energy added $= T\phi$.

In this the temperature is that at which the medium is receiving the energy but growing no hotter under the process as it is conducted at the fixed temperature of that source of heat which is supplying heat energy; and the entropy is that which belongs to the body after the increase of entropy ceases. It is the final or maximum state of the entropy, at the point of highest heat energy then under consideration.

A diagram, therefore, on which the absolute temperatures shall be the vertical ordinates and the entropy factor shall be the horizontal abscissæ will have an area enclosed between the bounding co-ordinates and the curves embodying their relations at intermediate points, which will present graphically to the eye at once the magnitude of the heat

energy•supplied under differing conditions and with different heat media.

When it is remembered that the adiabatic change of volume is that in which no change in entropy occurs, or the entropy is constant, it will appear that the changes in heat energy in adiabatic expansion or compression find convenient analogies in the descent of a weight of water W through a height measured from the sea-level of absolute zero down to a second level lower than the first, but still having a value greater than zero (cf. § 124). The available energy of the weight is thus

$$W(h_1 - h_2) = \text{available foot-pounds.}$$

By analogy, if the constant entropy during adiabatic expansion descend from an initial height T_1 to a final height T_2 on the absolute scale,

$$\phi(T_1 - T_2) = \text{available energy in heat-units,}$$

if a pound of the medium is under consideration.

Interesting extensions of this will be noted hereafter. It is further to be noted that the same ignorance as to the real meaning of weight or the attraction of the earth for bodies to it prevails for W in the hydraulic analogy as exists for the quantity ϕ in the heat energy diagrams.

197. Temperature-entropy Diagram for an Ideal Heat-engine.—An ideal heat-engine, as will be shown in a succeeding chapter, is one in which a given mass or weight of a heat medium is acted upon by heat to produce work, and returns after the completion of one cycle of operations to its initial state, the heat being supplied at a given constant temperature, and withdrawn or rejected at another constant temperature. The difference between the two quantities measures the work done in a perfect engine. The heat is therefore necessarily

supplied and withdrawn under the isothermal law for cónstant
temperature, with change in entropy; the expenditure of
energy on expansion is most effectively done by adiabatic
expansion, and the rise to initial state of entropy is also to be
done without change of such entropy, or by a second adia-
batic.   The pv diagram will therefore show a pair of adia-
batic curves (Fig. 79) connected at their extremities by a pair
of isothermals.   To express this set of relations on the tem-
perature-entropy diagram, with T ordinates and entropy
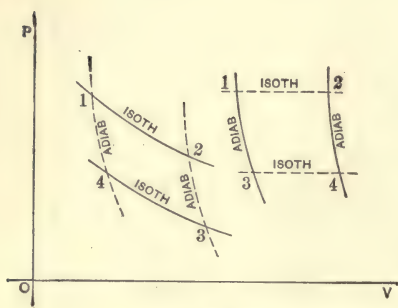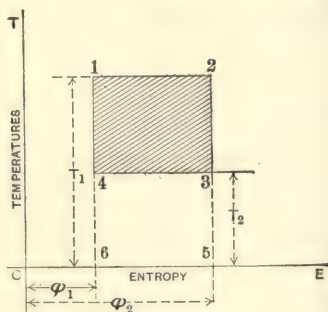
FIG. 79.   FIG. 80.

abscissæ, the isothermal line of increasing energy starts from
the upper end of the T ordinate corresponding to the temper-
ature (Fig. 80).   The top horizontal line 1, 2, has a length
representing the increase of entropy during that isothermal
process.   At 2 the adiabatic must begin.   The expansion
along an adiabatic is accompanied by no change in entropy,
but by a drop in absolute temperature to the point 3 corre-
sponding to T_2.   At 3 the withdrawal of heat and a lowering
of entropy occurs till the point 4 is reached, where the adia-
batic begins, with constant entropy and increase of tempera-
ture from T_2 back to T_1, closing the cycle.   The heat which
could not be utilized but was rejected is the area of the lower
unshaded rectangle represented by $T_2(\phi_2 - \phi_1)$, a quantity
which is obviously reduced as T_2 is made smaller, and be-

comes zero when T_2 is zero.   The relation between the entire
energy supplied or received by the medium

$$\text{area } 1256 = T_1(\phi_2 - \phi_1),$$

and the work done

$$\text{area } 1234 = (T_1 - T_2)(\phi_2 - \phi_1),$$

will be the efficiency of the cycle, or

$$\text{Efficiency} = \frac{(T_1 - T_2)(\phi_2 - \phi_1)}{T_1(\phi_2 - \phi_1)} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1},$$

a result to be otherwise deduced hereafter.

If the passage from 1 to 2 take place at 100 pounds pres-
sure of steam absolute, corresponding to a latent heat of 882
thermal units, and an absolute temperature of 788 degrees,
then the entropy counted from 32°, or freezing-point of water,
should be

$$\frac{882}{788} = 1.117,$$

for one pound of such water, and gives the length of the line
$\phi_2 - \phi_1$.

If the unit of heat medium was a permanent gas, then the
heat added during an expansion from v_1 to v_2 will be the
same as the work of such isothermal expansion, which was
found (§ 166) to be in foot-pounds

$$w_1 = p_1 v_1 \text{ Nap. log } \frac{v_2}{v_1}$$

$$= p_1 v_1 \text{ Nap. log } r$$

$$= JRT \text{ Nap. log } r.$$

If this value in foot-pounds be divided by J, it becomes heat-
units, and if the increase in energy be divided by T, at which

that increase took place, the quotient is the final value for ϕ; or (§ 124),

$$\phi = R \text{ Nap. log } r.$$

It is a matter of indifference at what distance from the point 6 the origin or zero of entropy be taken; or, in other words, what value of T_0 be taken as a datum. Temperature will of course be taken from absolute zero, if the value of heat rejected has any significance. Usually, however, it will be only the differences in temperature and in entropy which are required.

198. Deductions from the Temperature-entropy Diagram.—The use of the temperature-entropy diagram permits a direct and obvious deduction of certain facts and principles, also deducible from the pv equations but less simply.

(1) When heat passes from one body to another, the entropy of the system is increased. This follows because heat passes downward only from a warmer to a colder body (never automatically the other way); and if the heat-area transferred from the hotter and equal to the heat-area received by the lower have a higher and longer temperature-ordinate, the other area with lower and shorter temperature-ordinate must have a greater entropy value to give equality of area.

(2) Clausius announced the generalization that the entropy of the world tends to a maximum. This follows from the previous principle, since all transfers are downward unless mechanical force is introduced at the expenditure of heat to make them otherwise.

(3) The entropy imparted by adding heat so as to change the state of the heat medium from that belonging to one adiabatic to that belonging to another adiabatic is the same by whatever path the passage takes place. The diagram shows that the distance between the two parallel lines representing the adiabatics is everywhere the same (cf. § 124).

(4) The heat absorbed or given out by a heat medium in passing from one state to another is given by the area between

the curve which represents the change of state and two adia-
batics, one drawn through each extremity of this curve.

(5) If a series of equidistant isothermals be drawn be-
tween two adiabatics, they will cut off equal areas; or isother-
mals equidistant in temperature divide the heat into equal
parts.

(6) If two bodies differ sufficiently in heat energy, a part
of that excess of energy in the hotter body can be trans-
formed into mechanical work by a proper heat-engine, and
the remainder transferred to the cooler body. The test that
all available heat energy has been transferred is that the
entropy of the system has not been increased by a mere
transfer as given in (1). If there has been such a transfer,
the lost work is proportional to that increase in entropy.

(7) The temperature condition of a medium which is to
operate adiabatically in a heat-engine is a measure of its avail-
ability, since the lower temperature limit is fixed by that of
the coldest available body. The area of the diagram in-
creases as T_1 increases, when T_2 is fixed by the temperature
of water available for condensation. Hence:

(8) Where a given heat-energy is under consideration, in-
crease of entropy is concurrent with a loss of availability of
that energy.

(9) If the conditions of the preceding paragraph (197) be
applied to one pound of steam, within very small variations
of pressure, and consequently a small corresponding range of
temperature, the height of the figure on the pv plane becomes
also very small, or may differ from a rectangle by only an
inappreciable quantity. If the symbol v denote the volume
of the one pound of steam, the area on the pv diagram will
be $(p_1 - p_2)v$. The area on the $T\phi$ diagram will be
$(T_1 - T_2)\phi$, the temperatures belonging to the pressures of
the pv diagram. Hence

$$(p_1 - p_2)v = (T_1 - T_2)\phi, \quad \text{or} \quad \phi = v\frac{p_1 - p_2}{T_1 - T_2},$$

giving, when the relation of pressures and temperatures are given by a table, a method for calculating the value of ϕ.

(10) Since the heat energy which disappears on making a liquid at a necessary temperature and pressure into a gas at that pressure and temperature is an isothermal absorption of heat, the so-called latent heat can be expressed by the equation $L = T\phi$, since the entropy is the quotient of the applied heat divided by the temperature at which it was applied. Hence from section (9)

$$L = Tv\frac{p_1 - p_2}{T_1 - T_2}.$$

199. Entropy-temperature Diagram Applied to a Perfect Steam-engine, with Complete Expansion.—The perfect engine and the diagram of § 197 assumed the water to be at the boiling-point, so that in forming steam at the temperature T_1 there was only the latent heat to be added. The more usual case is that at which the water is condensed after expansion to a temperature T_2 by an isothermal process, and is returned as water at that temperature to the boiler as the source of heat. The water therefore requires to be heated from T_2 to T_1 as feed-water and then to be made steam and to receive entropy at the higher temperature.

For any temperature T, the entropy counted from an assumed origin at T_0 will be

$$\text{Entropy} = \int_{T_0}^{T}\frac{dH}{T},$$

which, if the specific heat of water be considered as constant and equal to unity (§ 143) for the limits in question, transforms into the equation

$$\text{Entropy} = \int_{T_0}^{T}\frac{dT}{T} = \text{Hyp. log } T - \text{Hyp. log } T_0.$$

the temperature T being for any state intermediate between T_1 and T_2. The first element of the diagram will therefore be a logarithmic curve through the points a and e (Fig. 81)
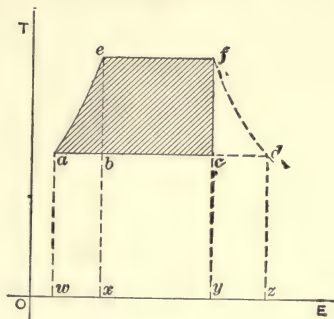
Fig. 81.

which have for their temperature-ordinate T_2 and T_1, respectively, and for their horizontal abscissæ

$$\phi_a = \text{Hyp. log } T_2 - \text{Hyp. log } T_0,$$

and for e,

$$\phi_e = \text{Hyp. log } T_1 - \text{Hyp. log } T_0.$$

This will give the distance

$$wx = \phi_e - \phi_a = \text{Hyp. log } T_1 - \text{Hyp. log } T_2.$$

It is not significant where the origin of entropy be taken, although in the figure the usual convention is observed, of calling the entropy of water at 32° F., zero. The weight of one pound of water is the mass of heat medium in question; T_1 corresponds to 103° F., or the temperature at one pound absolute pressure, and T_2 corresponds to 373° F., or the temperature corresponding to 180 pounds absolute. T_1 is therefore 834 and $T_2 = 562$. At the point e steam forms isothermally, and the length $ef = \dfrac{L}{T_1}$, if the entire pound of

water passes into steam. If a percentage x only becomes steam, the length will not be ef, but will equal $\dfrac{xL_1}{T_1}$, less than that belonging to complete vaporization. At f the cut-off takes place, and adiabatic expansion without change of entropy reduces the temperature down to T_2, when the exhaust opens and the entropy is reduced by the isothermal condensation process, at T_2 constant, which brings the cycle to the starting-point.

Examining now the heat interchanges:

Area $waex$ = heat taken in to heat feed-water to boiling-point;
    " $xefy$ = heat taken in during evaporation;
    " $wacy$ = heat rejected at exhaust;
    " $aefca$ = work done.

The rectangle $xefy$ is the analogue to the area in the previous paragraph, where the proportion of utilized heat to the heat applied was given by the ratio $\dfrac{T_1 - T_2}{T_2}$. The area to the left of this rectangle is the heat taken in during warming of the cool feed-water, and the utilized part abe bears a less ratio to the whole heat supplied, or the heat is used less efficiently. This is because the heat is not supplied at a constant temperature T_1, but at a temperature gradually changing from the lower to the higher value. That is, while this engine does more work and receives more heat than the engine of the previous paragraph, the work it does requires an amount of heat more than larger in proportion.

200. Amount of Condensation in Adiabatic Expansion. —The temperature-entropy diagram can easily be extended so as to give graphically the proportion of steam and water in an adiabatic process of expansion at any stage. If a curve cf be drawn from c (Fig. 82), whose points are found by drawing horizontals (isothermals) from the logarithmic curve ab, each

such horizontal being equal to the value of $\dfrac{L}{T}$ for that partic-
ular value of T, a curve will result concave outward, because
the entropy will increase as evaporation takes place at a lower
value for T. Hence if af be equal to $\dfrac{L_2}{T_2}$, and the steam were
entirely dry when it had cooled by expansion down to T_2,
the heat disposed of during condensation would have been
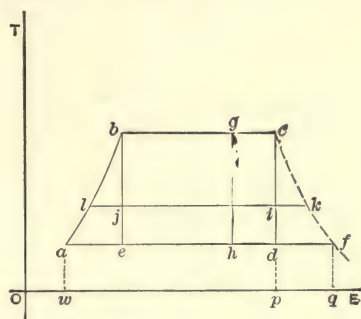
FIG. 82.

$qfaw$. But it was less than this actually, or the area $pdaw$.
Hence there must have been some part of the unit weight of
water already condensed before the temperature T_2 was
reached, or the length

$$\frac{ad}{af} = x, \quad \text{the percentage of steam, and}$$

$$\frac{df}{af} = 1 - x, \quad \text{the percentage of water.}$$

Similarly, any horizontal line $\dfrac{L}{T}$ between the curves ab and
cf will be divided by the adiabatic cd into segments giving
the proportion of steam at that temperature. For example,
at the point i, $x = \dfrac{li}{lk}$.

On the compression curve of the pv cycle, it will be apparent that the condensation should be stopped at a point e on the line fa if by compression of the heat medium it is to be brought to the temperature T_1 as water ready to evaporate. In other words, if adiabatic compression is to be used to restore the medium to the condition T_b, the process must begin when $\dfrac{ae}{af}$ represents a proportion of steam still uncondensed, and at any point of the adiabatic compression eb the dryness is $\dfrac{lj}{lk}$.

Finally, if the entire pound of steam is not dry vapor when the expansion begins, but only a proportion x. Let the point g give the proportion of complete vaporization which has taken place, so that

$$bg = \frac{xL_1}{T_1},$$

$$x = \frac{bg}{bc}.$$

The perpendicular through g now gives the line of complete vapor adiabatic expansion, and the segments to the right as before are water. At the exhaust period, $x = \dfrac{ah}{af}$, and the proportion of water is $\dfrac{hf}{af}$.

201. Temperature-entropy Diagram when the Expansion is Incomplete.—If the steam or heat medium expands down to the temperature and pressure represented by the exhaust at T_2, it must follow that at the end of the piston-traverse there is little or no forward effort acting upon it. This tends towards irregular motion, which must be counteracted by stored kinetic energy in fly-wheel or the reciprocating masses of the mechanism, or else the effort must be main-

tained by causing the motor-pressure to drop to T_2 only after the stroke has been completed. This is a practical condition which prevails very widely, and will result in changing the heat-diagram by a loss of availability along the lower line. In Fig. 83 let the expansion take place from the point of cut-off c adiabatically until a temperature is reached at the end of the piston-traverse corresponding to the point c' in the figure, at which pressure and temperature (above T_2) the exhaust opens and

FIG. 83.

available heat is swept out by a non-adiabatic expansion and condensation, resulting in a fall both in entropy and temperature until the line af is reached proper for T_2, whence the return stroke da brings the substance back to the starting-point.
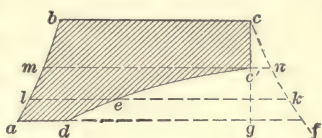
The curve from c' to d represents the change at the end of the stroke, before the piston reverses its motion, and is a curve of constant volume, while varying in pressure and temperature from the withdrawal of heat energy by condensation from contact with some cooler body. It is described by points. Any point on it will be found at the intersection of the temperature ordinate for T with the line drawn from the curve ab, which has a length representing the percentage of steam in the mixture at that temperature. The length le is to be to the length lk as the original volume xV' (in which V' is the volume of one pound of saturated steam at the temperature corresponding to the point c', but which must have moisture in it represented by $c'n$, so that $x = \dfrac{mc'}{mn}$) is to the volume belonging to the assumed temperature T. Or, in other words,

$$le : lk :: xV' : V, \quad \text{or} \quad le = \frac{xV'}{V}lk.$$

The loss of available energy as compared with complete expansion is the area bounded by the lines $c'g$ and dg and the curve $c'eg$. In the scale used in Fig. 83, the drop in temperature represented by the line cc' is from the pressure of 180 pounds to 3 pounds pressure.

202. Temperature-entropy Diagram when there is no Expansion.—If the steam follows at full pressure to the end of the piston-traverse, and is there released at full pressure to exhaust without any adiabatic expansion, there is no length
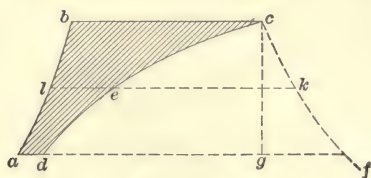
Fig. 84.

cc' in the diagram, but the curve of constant volume of saturated steam as temperature is lowered starts from the corner c (Fig. 84). Points on the curve are found as before, since $xV = V_1$, and hence

$$le : lk :: V : V_1, \quad \text{or} \quad le = \frac{V_1}{V} lk.$$

The area of lost work is the area cgd, which is greater than in the preceding case, the conditions being assumed the same, or T_1 as 834° and T_2 as 603° absolute. The line da, as before, shows the change in state or the condensation occurring as the piston returns.

203. Temperature-entropy Diagram when Steam is Superheated.—If the specific heat of steam undergoing a process of heating beyond the saturation point be represented by the factor 0.480 (§ 146), then the entropy will be increased after the point c is reached by an amount which will exceed that at T_1 by the quantity

$$\phi_s = 0.480 \int_{T_1}^{T'} \frac{dT}{T} = 0.480 \, (\text{Hyp. log } T' - \text{Hyp. log } T_1).$$

The effect of this upon the diagram is to add a curve plotted
by points as given for a series of values of T' up to the max-
imum temperature used.   The curve 3–7 is such a logarithmic
curve (Fig. 85), and the point 7 indicates
that maximum temperature.   At this
point, if adiabatic expansion sets in, the
line 7–4 is described with complete expan-
sion.   The heat taken in is increased by
3 7 9 8, while the work done is increased
by 5 3 7 4 as the result of the superheating.
It will appear that even if the superheat-
ing be considerable (200° in the scale
selected) the proportionate increase is
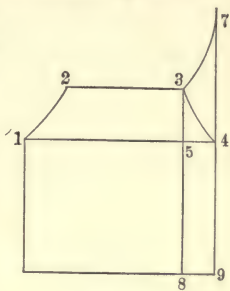
FIG. 85.

small, as compared with the total expenditure.   In other
words, so much of the heat is taken in at the temperature of
saturation that the subsequent raising of temperature, even
to a considerable degree, offers small theoretical advantage.
It offers a practical advantage, however, as will be shown
hereafter (§ 229.   See also § 232).   The temperature may
drop in expanding down to a point on 3–4, where the adiabatic
line would cross the saturation curve before condensation
begins.   Up to this point the steam has been superheated.
If the steam were to be dry at the end of expansion down
to T_2 then the curve 3–7 would have to be prolonged until
it met the temperature ordinate through 4.

**204. Plotting of Entropy-temperature Curves for Water
and Steam.**—For the convenient use of the entropy-temper-
ature diagram, the logarithmic curves for the relation of tem-
perature to entropy may be conveniently plotted on cross-sec-
tion paper, and on the scale preferred for the heat diagrams,
and then used directly between the desired limits in drawing
in the curves of heating of water and in the heating of the
steam-gas.   As drawn in Fig. 86, the horizontal distance
between the two curves gives the change in entropy which
takes place when water at any temperature is changing into

steam at the same temperature $\left(\dfrac{L}{T}\right)$.   The numerical values
start at zero at 32° F., and are for one pound weight.   The
specific heat of water increases as the temperatures increase,
it will be remembered (§ 139).
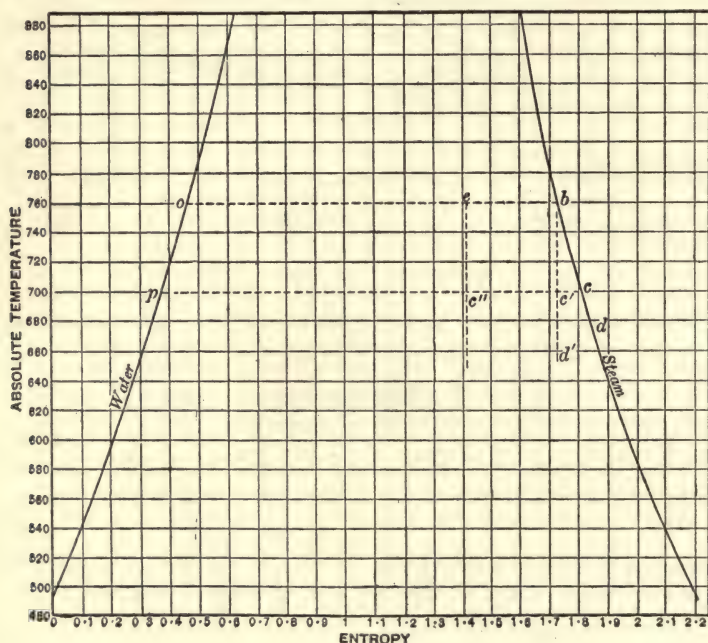
FIG. 86.

This pair of curves is also of great convenience in drawing
in the curve of saturation for an adiabatic expansion in terms
of pressure and volume.   If the curve BCD (Fig. 87) repre-
sent a curve plotted from the tables for saturated steam, let
it be required to draw the adiabatic through B.   Let C be
any point on the saturation curve whose pressure is given,
and for which tables give the corresponding temperature.
From the point b on the entropy curve drop a perpendicular
(a line of constant entropy) to meet the line pc drawn through
the temperature point corresponding to C.   This will give the

point c'. Then the point C' on the adiabatic curve will lie at a distance NC' from the line of no volume given by the relation:

$$NC' : NC :: pc' : pc.$$

Other points are similarly found. If the initial volume in cubic feet were the result of vaporization of less than one
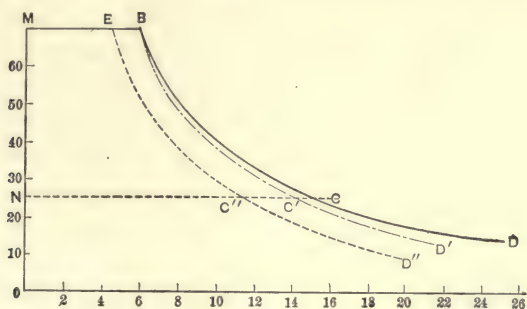
FIG. 87.

pound of water, or the percentage of steam was less than 100 at the beginning, then the curve of vaporization of such a mixture, as represented by $\dfrac{ME}{MB}$ per cent of steam and $\dfrac{EB}{MB}$ percentage of water could be found by dividing the space ob in Fig. 86 in that proportion, and using the ratio given by the line ec''. Points on the adiabatic through E would be found by making

$$NC'' : NC :: pc'' : pc.$$

This same diagram can be used to find the curve cf in the foregoing sections.

It will be apparent from the slope of the entropy curves that for each range of temperature there is a certain adiabatic for which the initial and final figures are the same. If the steam contains more than 50 per cent of water within the range given, it will become drier by expansion; if less than this, it will become wetter.

205. Transfer of the Indicator-diagram to the Temperature-entropy Diagram.—The foregoing paragraphs have made clear the methods to be used with any actual case in transferring the pv points on the curve traced by a steam-engine indicator-pencil to the temperature-entropy diagram-plane. The theoretical diagram is first drawn, giving the ideal condition of complete vaporization, with the expansion curve that of saturation, and with cut-off located with respect to the line of zero-volume, thus neglecting clearance. The corresponding ideal temperature-entropy diagram is then drawn, with values for T_1 and T_2 corresponding to the given pressures, and the entropy values laid off according to the observed data as taken from tables. The actual engine-diagram will lie within the theoretical one in both cases, by reason of the losses in pressure and temperature caused by the necessity of operating engines in air cooler than the medium, and for other causes to be discussed in a following chapter. Hence it is the problem to locate points within the theoretical pv diagram at the corresponding points within the theoretical te diagram. The vertical distances above the zero of temperature on the te diagram are taken from steam tables which give the temperature corresponding to the pressure on the pv curve. To locate the upper end of these pressure-temperature ordinates on the entropy scale, the principle is used that corresponding points in each actual diagram divide the horizontal lines of the theroretical diagrams in the same proportion. That is, if X in Fig. 88 be a point in the actual indicator-diagram, a horizontal AB is drawn through X on the pv theoretical diagram, and also a line ab through the point on the te diagram which is at the distance from the line of zero temperature required for the pressure corresponding to X on the other. Then divide ab in the same proportion as the point X divides the line AB, which determines the desired point x. It will be apparent that if the volume of the liquid be not considered, the percentage of

liquid evaporated when X denotes its condition will be in the same ratio to AB, denoting complete evaporation, as the heat required for this evaporation will be to the heat required for such complete evaporation.   Or,

$$AX : AB :: ax : ab.$$

This can also be worked out from entropy tables or from the diagram of § 204.   All the points of the indicator-diagram can thus be located, and an area enclosed representing the heat energy attaching to the work in foot-pounds of each pound of steam used, and its departures from the ideal con-
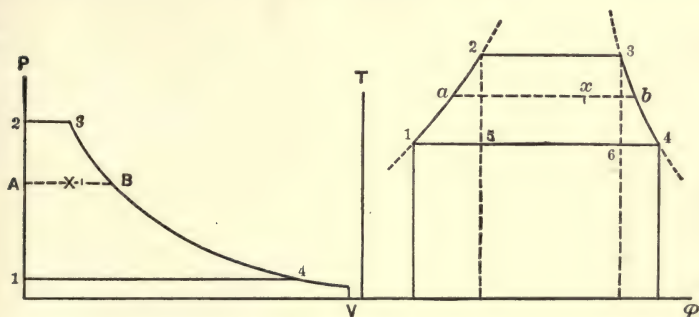
Fig. 88.

sumption measure the efficiency of its operation.   Detailed examples will be referred to hereafter when certain other points have been discussed.   Its exact application involves that certain desirable refinements should be made more clear.

CHAPTER XV.

THE IDEAL CYCLE HEAT-ENGINE.

210. Introductory.—In the discussions of the preceding chapters certain principles have been assumed by implication, to which it is desirable to refer now more in detail.

It has been made obvious by the heat diagrams of the preceding chapter that heat energy is made available for mechanical work by the existence of a difference in heat level between bodies. Just as no work could be done by water-power if all water were at a dead-level of the seas, so no mechanical effect could be produced, however great the amount of actual heat energy, if all bodies were at a dead-level of temperature. Hence it is desirable to separate T_1 and T_2 by as great an interval as possible, in heat-engines.

Secondly, the lowest available temperature will be fixed by the temperature pertaining to that climate or latitude which attaches to the best cooling medium there. This is usually water, by reason of its high specific heat, and it rarely can be counted as having for the year round a temperature as low as $50°$ F. in the temperate zone. Hence, it will be impossible to convert the whole of any heat supply into work, because the temperature T_2 thus fixed is so far above the absolute zero of temperatures that a considerable quantity of heat must always be unavailable, and will be swept out by the exhaust.

Thirdly, if the highest temperature be a temperature T_1, and the lowest practicable temperature be T_2, it is obvious that any heat taken in below T_1 will have less availability for

conversion into work than if it had been taken in at T_1, or will be less effective. Similarly, it will entail a loss if any heat is rejected above a temperature which corresponds to T_2.

Fourthly, complete expansion will be more efficient than incomplete expansion, and should be so managed that there are no losses from eddies or internal movement in the medium such as will occur with a free expansion or one which is imperfectly resisted by the external resistance being overcome.

It was upon the basis of these assumptions and deductions that Sadi Carnot in 1824 proposed the classic conception of a heat-engine whose heat medium should traverse a succession of *pv* relations between two fixed limits of temperature, and return to its initial state after each traverse. The complete path he called a *cycle*. He applied it first to a permanent gas, used as a heat medium. The Carnot cycle, and the first and second laws of thermodynamics, need to be noted.

211. The First Law of Thermodynamics.—The science of thermodynamics means by definition the science of heat energy. Its first and fundamental law has already been enunciated (§ 9), that heat and mechanical energy are mutually convertible; and that heat requires for its production the expenditure of a definite number of units of work, or by the expenditure of heat a definite number of units of work can be done. It is obvious that back of this law lies the fundamental conception of the conservation of energy, which may be stated: '' Energy cannot be created nor can it be annihilated by any physical processes which the mind can conceive.''

212. The Second Law of Thermodynamics.—What is known as the second law of thermodynamics has been variously enunciated by the great masters who have studied it. The axiom of Clausius may be treated as a first section of the second law: ''Heat cannot pass of itself from a colder body

to a hotter one," meaning that of two bodies, that to which heat energy passes is always the colder one, unless mechanical energy comes in from without. This is otherwise expressed by saying that "A self-acting machine cannot convey heat from one body to another at a higher temperature."

Rankine's statement of the second law may be combined with the foregoing: " If the absolute temperature of any uniformly hot substance be divided into any number of equal parts, the effect of each of those parts in causing mechanical work is equal." That is, provided the work of transfer is done in the most efficient way, the equal intervals into which any range of temperature may be divided are equally effective when heat is allowed to pass through all the intervals from the top to the bottom of the range. This is graphically obvious from the temperature-entropy diagram of Fig. 80. It assumes, therefore, in it the conception of the absolute scale, and that Carnot's cycle is used in effecting the transfers of heat into work.

213. Carnot's Cycle.—Carnot's cycle is the realization of the expansion, compression, and heating and cooling of a perfect gas under conditions which § 197 has shown to be those of maximum efficiency. The heating and cooling must be done at a constant temperature, and therefore these changes of pressure and volume in the working cylinder must be by the isothermal law to secure maximum effect. The expansion must be done without additional heat supply and without loss externally; it must therefore be adiabatic. The compression similarly must be without change in entropy, but only cause a change from the temperature T_2 back to T_1; the compression must therefore be adiabatic. The conditions, therefore, imposed by the Carnot cycle are those presented in Fig. 89. The cylinder and piston must have no heat capacity, nor friction. The bottom of the cylinder is a perfect conductor. The element A is a heat-source of great capacity at a temperature T_1, and the element C is a condenser of great

capacity maintained at T_2. Both the source of heat and the cold body are so capacious that no change in T_1 or T_2 can occur while the pound of gas behind the piston is heated by contact with the heater, and cooled by contact with C. B is a non-conducting cylinder-cover, for use when the perfect gas is expanding adiabatically and without influence of heat and
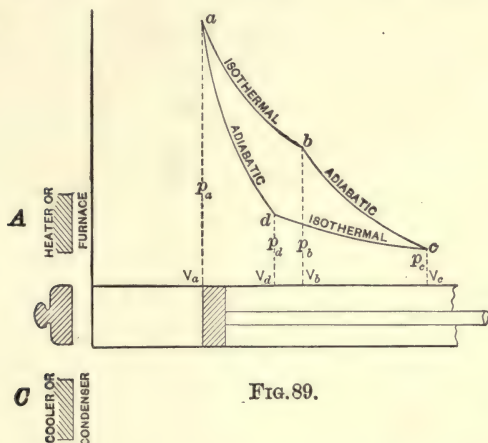
Fig. 89.

cold. The relations of pressure-volume for the various stages are given by the subscripts on the diagram. The specific heat at constant pressure will be denoted by C_p and the ratio between the initial and final volumes $\dfrac{V_c}{V_d}$, which must be the same as the ratio $\dfrac{V_b}{V_a}$, will be denoted by the factor r, or the ratio of the expansion.

There will be four steps or stages. The student-reader is advised to compare the procedure given in §§ 197–198:

(a) Apply the heater A. The piston rises; the unit weight of gas expands isothermally at T_1. The heat energy taken in is

$$H_1 = CT_1 \text{ hyp. log } r,$$

which all goes to increase entropy.

(*b*) Heater A is removed, cover B is applied, and the piston moves out at the expense of its temperature, without change of entropy, until the temperature falls to T_2 by such expansion against external resistance.

(*c*) Take away cover B and apply cool body or condenser C at T_2. No change will take place, because the expansion is complete, unless the piston be pushed back. But if the piston is retracted the smallest tendency to an increase of temperature above T_2 is at once met by a flow of energy into the condenser. The gas loses entropy down to the stage represented by T_2, and the amount rejected to the condenser will be the difference between the entropy as T_1 and T_2, or

$$H_2 = CT_2 \text{ hyp. log } r.$$

(*d*) Remove the condenser C and replace B when the point d is reached. The determination of this point has already been touched upon and found graphically (§ 199). The piston is now still further forced in and back until the gas has its initial volume V_a, and if the point d was rightly chosen it has also the temperature T_1 at which it started because the compression has been adiabatic, and the cycle has been completed. For the relations of V_b and V_c to produce the desired final temperature T_2, refer to § 168, which will give.

$$\frac{T_1}{T_2} = \left(\frac{V_c}{V_b}\right)^{n-1} \quad \text{and} \quad \frac{T_1}{T_2} = \left(\frac{V_d}{V_a}\right)^{n-1}.$$

according as the location of b or d is desired.

It will appear, therefore, that the Carnot cycle gives an external work in foot-pounds which will be 778 times the difference between the heat rejected and the heat received, or, for the complete cycle,

$$\text{Work} = 778C(T_1 - T_2) \text{ hyp. log } r.$$

which is 778 times the area included in the diagram of curves (Fig. 89), all transfers having been made at maximum efficiency.

214. Carnot's Cycle Reversed.—In the foregoing conception, the curves have been described by starting at a and following round the cycle clockwise, as it were. If, however, the start be made from a and the curves described in the reverse order, the condition of affairs is that in which mechanical energy is converted into heat, or Carnot's cycle is reversed. What happens then will be:

(e) The cover B being in place, the piston is drawn to the right till the point d is reached. The adiabatic ad is traced, and the gas cools down to T_2.

(f) Cover B is removed, and condenser C is applied. The piston is drawn out still further to the right, but as the gas is in contact with C at the temperature T_2 it cannot fall below that temperature in expanding, and heat flows from the condenser according to the isothermal law, to the amount

$$H_3 = C_v T_2 \text{ hyp. log } r.$$

(g) The point c being reached, the condenser is detached, the non-conducting cover B is replaced, and by external mechanical energy the piston is forced back to b. The compression being adiabatic, the temperature rises to T_1 without rejection of heat in the process.

(h) Further compression back to initial volume with cover B removed, and the heater A applied. Heat flows into A because the compression must be isothermal, the curve ba is described and the heat which passes into A will be

$$H_4 = C_v T_1 \text{ hyp. log } r.$$

It will appear from a comparison of H_1 and H_4 and of H_2 and H_3 that these are equal, or the same amount has been put back into A by the reversed process as was taken out in the direct cycle, and the same amount was taken out of C in the reverse process which was rejected into it on the direct cycle.

A cycle capable of being operated in either direction is called a *reversible cycle*.

215. Carnot's Criterion of Reversibility.—It is not diffi-
cult to show that the efficiency of the reversible engine is a
maximum.   Let it be conceived that there are two heat-
engines working between the same limits T_1 and T_2, one non-
reversible operating by the direct cycle, to be designated by
D, and the other a reversible engine operating on the reverse
cycle R.   Suppose both to be capable of being connected to
the same heater A and the same condenser C and to with-
draw and apply heat as they operate.   Let them be sup-
posed equally frictionless, and that they are connected to-
gether so that D drives R, as a steam-engine drives an air-
compressor, without loss.   Then if the two machines were
equally efficient, the heat withdrawn from A to drive D would
be restored by the pumping action of R, and the heat added
to C by D would be withdrawn by R.   The result would be
an indefinite continuance of the action, without addition of
outside heat or mechanical energy.   If, however, R is more
efficient than D, and puts back into A more heat than D has
withdrawn from it in order to drive R, then it will follow that
the cold body from which R derives its heat is transferring
heat to a warmer body, which is contrary to all experience,
and to the enunciation of that experience in the second law
of thermodynamics.   But D is *any* engine working between
the same limits of T_1 and T_2.   Hence no engine can be more
efficient than the reversible one.   But may not the direct
engine D be less efficient than the reversible engine?   This is
met by assuming both engines to be reversible, and following
the same reasoning.   It is similarly proved that neither can
be more efficient than the other; whence:

(1) The reversible heat-engine has the maximum effi-
ciency when the limits T_1 and T_2 are given.

(2) All reversible heat-engines working between the same
limits of temperatures are equally efficient; that is, the effi-
ciency in the thermodynamic sense is independent of the heat
medium.

This second form of enunciation has been preferred by

some for one of the statements of the second law of ther-
modynamics.

216. Efficiency in the Carnot Cycle.—Remembering that
the efficiency of any machine is the ratio which the total avail-
able work bears to the work actually realized, the efficiency
of a Carnot engine will be:

$$\frac{\text{Heat utilized}}{\text{Heat applied}} = \frac{C(T_1 - T_2) \text{ hyp. log } r}{CT_1 \text{ hyp. log } r} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}.$$

But the foregoing discussion has shown that no reversible
heat-engine can be more efficient than another working
between the same temperature limits. Hence the above
expression for efficiency is that of maximum possible effi-
ciency, or is the measure of perfect efficiency for all engines
receiving and rejecting heat under the conditions assumed by
Carnot.

Most heat-engines, with the exception of the gas-engine,
operate under this assumption. In the steam-engine, for
instance, the capacity of the cylinder up to cut-off is filled by
vaporization at a constant pressure and temperature, and the
condensation or exhaust is at a lower constant pressure and
temperature; the expansion and compression are adiabatic,
as assumed by the Carnot requirements. The difficulty is,
however, that the actual engine does not and cannot reach the
ideal efficiency for reasons to be made apparent in the next
chapter, which are concerned with the actual construction of
the steam-engine and are not capable of being reduced to non-
experimental statements.

It will be observed in discussing the conclusion:

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}.$$

(1) The efficiency increases with higher temperatures (and
pressures) for the heat medium as it enters.

(2) The lower temperature being fixed by available cooling
bodies for condensers, the efficiency can never reach unity
while T_2 has to be so large. Figs. 90 and 91 show a plotting

CARNOT CYCLE. NON-CONDENSING STEAM ENGINE CYCLE
$p_2 = 15$

FIG. 90.

CARNOT CYCLE. CONDENSING STEAM ENGINE CYCLE
$p_2 = 2$
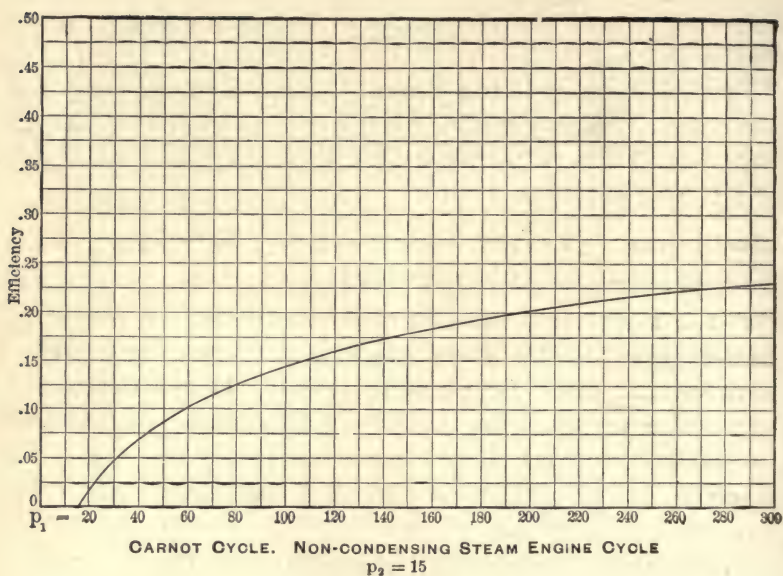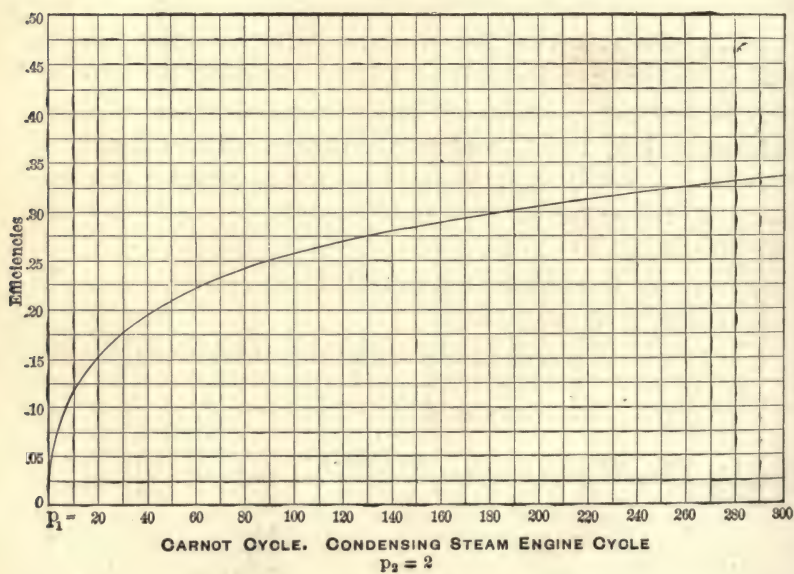
FIG. 91.

of such efficiencies with T_2 chosen as 212° F. in Fig. 90 and with T_2 as 125° for Fig. 91.

(3) The effect of condensing the steam is to lower T_2 and of compounding cylinders is to make it easier to raise T_1 (§§ 172–174).

(4) The efficiency does not include pressure or specific heat or other physical properties of the medium, but is dependent on temperature limits alone.

(5) Air as a heat medium, allowing a higher value for T_1 before the pressure becomes troublesome, is a more efficient heat medium than steam where this difficulty occurs. There must be some other things than heat efficiency to consider.

(6) The low efficiency of the steam-engine thermally is not a reflection upon its effectiveness as a means of transforming the *available* energy of a fuel into mechanical energy. The mechanical efficiency is not to be confused with the thermal efficiency. This latter is limited by the greatness of the minimum value for T_2.

217. Rankine and Clausius Cycles.—The Carnot cycle for the steam-engine involves the condition, which is not usually realizable, that the heat medium is raised in temperature by compression. Rankine introduced a cycle and elaborated formulæ for its efficiency in which the succession of curves are of an ideal indicator-diagram. The isothermal is horizontal for the period of admission; the expansion is either an adiabatic or a saturation curve; the isothermal for the back-pressure line is either at the level of complete expansion or below it; and finally the effect of the water-volume in the steam and any effect of entrapped steam used as a cushion in compression are neglected. This cycle, like the Carnot, is approachable, but not attainable in practice. It offers the advantage that in the design and test of an actual engine it may be compared with an ideal one which may be called a purely thermodynamic machine of similar construction and like limits of pressure, temperature, and degree of expansion.

A cycle by Clausius is the Carnot cycle without compression, but with complete expansion down to the back-pressure line. It permits the measure of the loss incurred when the Rankine cycle is used without such complete expansion.

218. Theoretical Weight of Heat Medium for a given Work.—An interesting practical deduction from the Carnot equation for efficiency is easily made. Since

$$\frac{T_1 - T_2}{T_1} = \frac{\text{Heat utilized}}{\text{Heat applied}},$$

the second member can be reduced to foot-pounds by multiplying both terms by 778. Hence

$$\frac{T_1 - T_2}{T_1} = \frac{\text{Work utilized in foot-pounds}}{778Q_1},$$

in which Q_1 is the product of the weight of heat medium into the heat which it has absorbed in reaching the temperature T_1. In the case of steam, this heat will be the heat of vaporization, or the latent heat, or the product T_ϕ, when the feed-water at the temperature corresponding to the pressure T_1 is made into steam at that pressure. In § 139 this was designated by r, and is found by subtracting the heat of the liquid (q) from the total heat (λ). Hence it is only necessary to substitute for the work term the number of foot-pounds of work corresponding to a horse-power per hour ($60 \times 33,000 = 1,980,000$), and for Q_1 a factor made up of the unknown weight of heat medium sought multiplied by its heat of vaporization at the temperature T_1. That is, if M denote the desired weight of heat medium,

$$Q_1 = Mr_1.$$

Whence

$$\frac{T_1 - T_2}{T_1} = \frac{1980000}{778Mr_1},$$

whence

$$M = \frac{T_1}{T_1 - T_2} \times \frac{2545}{r}.$$

It is interesting to note that the weight of heat medium decreases as the heat of vaporization increases, and that the latter is the measure of the amount of work which will be done by a unit of weight of the medium. The factor 2545 is interesting as presenting the number of units of heat to be converted per hour into work for each horse-power.

A table of efficiencies and theoretical water-consumption per horse-power may be computed on proper assumptions for condensing and non-condensing engines as follows :

WATER CONSUMPTION AND EFFICIENCY.

Gauge Pressure above Atmosphere. Initial.	Condensing.		Non-condensing.	
	Efficiency. $\frac{T_1 - T_2}{T_1}$.	Pounds of Steam per H. P. per Hour.	Efficiency. $\frac{T_1 - T_2}{T_1}$.	Pounds of Steam per H. P. per Hour.
15	0.189	14.3	0.053	50.9
30	0.215	12.8	0.084	32.8
60	0.249	11.4	0.124	22.9
100	0.278	10.5	0.157	18.4
150	0.302	9.8	0.186	16.0
200	0.320	9.5	0.207	14.6
300	0.347	9.0	0.238	13.1

The temperature t_2 is 212° F. for non-condensing engines, and for the condensing engines it was made 115°, corresponding to 1.5 pounds pressure.

In engines using a permanent gas like air, which has no heat of vaporization, as in the case of vapors, the factor r is replaced by latent heat of expansion, which is the product of the entropy by the temperature.

The same result can be secured from the temperature-entropy diagram directly. Assuming the diagram of Fig. 80,

which presents the Carnot cycle of maximum efficiency, it is apparent that the temperature-entropy area in heat-units can be made the area in foot-pounds by multiplying both members by 778. That is,

Area in heat-units $= (\phi_2 - \phi_1)(T_1 - T_2).$

Area of work in foot-pounds $= 778(\phi_2 - \phi_1)(T_1 - T_2).$

But the factor $\phi_2 - \phi_1$ is the change in entropy at T_1 in passing from water at T_1 to steam at T_2, which is $\dfrac{r_1}{T_1}$ for one pound of fluid and becomes $\dfrac{Mr_1}{T_1}$ if an unknown weight is to do the work imposed by giving a value to the foot-pounds of the first member. Hence

Water per H.P. per hour $= 778\dfrac{Mr_1}{T_1}(T_1 - T_2).$

Whence

$$M = 2545\,\frac{T_1}{r_1(T_1 - T_2)},$$

as before.

In the case of a permanent gas, the value for $(\phi_2 - \phi_1)$ for an isothermal expansion was found to be

$$R \text{ hyp. log } r.$$

Whence the equations become

Work of one H.P. per hour $= 778MR \text{ hyp. log } r(T_1 - T_2),$

whence

$$M = \frac{2545}{R \text{ hyp. log } r(T_1 - T_2)},$$

in which r is the ratio of the final to the initial volumes in the expansion process.

219. Recapitulation.—The Carnot cycle being the cycle within which must lie the performance of actual engines, and to which they should approximate as closely as possible to make the actual value of M small and to make the expenditure of fuel to raise it to T_1 as small as possible, it becomes of interest to examine the causes of difference between the ideal and the real engine, topics which form the next chapter. But it should not be overlooked as a fundamental departure from actual conditions, that the Carnot cycle for the steam-engine demands three impossibilities:

(1) That the body of water in the boiler be always at the temperature T_1 no matter what the exigencies of feeding, or what feed-water temperature be available, and that there be no drop of temperature in supplying the cylinder.

(2) That all heat be rejected from the cylinder at the lower temperature, T_2, and not by a process of gradual cooling. That is, the steam in giving up its heat shall be at the temperature of the condenser; if it were, it would not give it up.

(3) That all heat delivered to the medium shall be carried down from T_1 to T_2 purely adiabatically, without being diverted by radiation or contact or other methods of transfer, in spite of the conducting qualities of the structural materials used, in spite of eddies in the steam itself, and in spite of possible free or unresisted expansion, "drop" into the lower pressure of condensing appliances, receiver, and the like.

Finally, the thermal and not the mechanical character of the efficiency equation needs to be emphasized, and its application only to any one medium to which it is for the moment applied. A study of the temperature-entropy diagram, however, for a Carnot cycle, shows that when the medium changes and another is used which has a different entropy value at T_1, the efficiency ratio changes in the same proportion (another way of saying that the thermal efficiency depends on the temperature ratio alone); but the Carnot equation does not say that the same weight of different media will

be required with a same temperature range, nor that all
media have to be used at the same range, nor that the heat
to be expended to raise all media to T_1 is the same, nor the
extent of condensing appliances to cool these different media
to T_2.   This belongs to a different department of the subject,
and will be treated in Chapter XXI.

CHAPTER XVI.

THE CYCLE OF THE ACTUAL STEAM-ENGINE.

220. Introductory.—It has already been said that the actual engine must depart from the Carnot ideal because—

(1) Heat is not received from the furnace at a constant high temperature T_1.

(2) Heat is not rejected at a constant lower temperature T_2.

(3) All the steam-heat energy is not devoted to work.

221. Elements of Departure of the Actual Cycle from the Ideal Carnot Cycle.—But besides these, and belonging to a class which the engine-designer can control in part, are other sources of loss or ineffectiveness, to which attention must be called. Among these are:

(4) Loss of pressure and temperature from friction and radiation and conduction in the steam-pipe between the boiler and the engine.

(5) For this cause T_1 at the engine is not the same as at the boiler. The steam is therefore not dry, but carries a mist of watery particles resulting from condensation, and the entropy value is not that belonging to dry steam at T_1.

(6) In the engine itself, at throttle-valve, governor-valve (if any), and at the ports of the valve or valves by which distribution of steam is effected into and out of the cylinder, a loss of pressure occurs by the process known as "wire-drawing" from friction and the work of overcoming it.

(7) Condensation, and entropy-drop by contact of the hot

incoming steam with a cylinder-head and one side of the piston, and the steam-passages to that end of the cylinder, which have just ceased a contact with steam at the lower temperature T_2. The exhaust steam has cooled these surfaces off and swept out the heat with itself, and initial condensation covers these cooler surfaces with dew. This is an entropy-leakage of which theory can take account with difficulty, and which yet affects greatly the value of M (§ 218).

(8) Even if the cylinder-walls and cover were absolutely non-conducting and non-diathermanous, a condensation of steam will occur after cut-off and during the process which is alleged to be adiabatic. The conversion of heat into work must result under non-isothermal conditions in the condensation of a certain percentage of steam to water, or (unless steam-jacketed) the actual curve of the indicator-card will fall within the curve of saturation for steam as laid out from tabular values (§ 200). It happens, however, usually, that as this condensed percentage of moisture lying on the cylinder bottom or in the form of dew on the metallic surfaces is reduced in pressure by the increase in cylinder volume during expansion, the point is reached at which the equalization of temperature and boiling-point for water is also reached. When this occurs, if the cylinder-walls will furnish the necessary heat energy to supply entropy to this water, it will absorb the heat of vaporization r which it requires at this lowered pressure, and the steam formed will raise the pressure ordinate on the indicator-diagram, and the cylinder metal has been cooled still further. Here again the theoretical diagram gives no hint of this entropy reaction, but the incoming steam has to supply the new heat called for by an increase in the loss discussed under (7). The heat supplied to the steam by this re-evaporation of condensation is swept out at exhaust and is lost.

(9) At the end of expansion the exhaust opens, and the release occurs to a condenser or to the atmosphere. It has

been already noted that in steam-engines it is not usual to se-
cure complete expansion (§ 201) because the forward effort
becomes ineffective at the end of the stroke. If the valve-
gear is automatically adjusted by variation in the load, with-
out change in steam-pressure, it may easily happen that the
pressure at the release is rarely or never that of the back-pres-
sure or exhaust line. If above that point, there is the loss
from non-adiabatic expansion on the temperature-entropy
diagram (§ 201), and the exhaust sweeps heat out unutilized.
If the terminal pressure at release is less than that represent-
ing the back-pressure, the expansion curve has crossed the
back-pressure line, forming a loop at this end of the diagram.
The fly-wheel must therefore be making the engine overcome
its own friction, and be doing a little pumping action in the
cylinder; while the contents of the exhaust-passages will
evince a tendency to reverse their outward direction, at a cost
of mechanical energy, and the expanding steam loses by free
or unresisted expansion.

(10) During exhaust the pressure attaching to the actual
T_2 of the condenser or the atmosphere may not prevail in the
cylinder, by reason of friction or wire-drawing of the exhaust
outflow from valves, passages, and piping connections.

(11) During exhaust at or near T_2 the cylinder-walls and
piston-head are radiating heat to the exhausting volume of
saturated steam, containing also perhaps a mist of water un-
evaporated. This is a different heat from that which the
heat medium is giving up, but which must be supplied at the
next stroke by the incoming steam [see (7)].

(12) The exhaust-valve and port probably close before the
end of the return stroke, entrapping some steam and com-
pressing it adiabatically in the main, raising its pressure and
temperature as the volume diminishes. If prudently done,
the pressure may rise to the initial pressure. It is usually
done by the excessive living force of the reciprocating parts
which would otherwise be wasted, and the elastic steam-cush-

ion serves to lessen the flexing effect on the crank-pin which would otherwise receive it. The heat due to this compression is regained for the cylinder-walls, although under absolutely ideal conditions the heat represented by the energy of the reciprocating parts would not have been wasted upon them in the first place, but should all have been absorbed in useful work on the crank-pin. If compression is carried too far, a loop forms on the top of the compression curve, with free expansion losses, excess of pumping action, and arrest of steam-flow into the cylinder.

(13) The clearance space left between the piston and the cylinder-head at each end, to prevent impact, to lessen trouble from water, and to give a volume in which forward pressure may establish itself at dead-centres, adds a volume to the actual piston displacement caused by the stroke. The valve-passages add a further necessary waste volume, which must be filled with steam at full pressure and temperature at each stroke, and which is exhausted (less the compression or cushion steam) without having done as much work as if the expanding volume had been smaller. The heat to make this wasted volume of steam is lost.

(14) It is only in condensing engines that the exhausted steam after condensation is pumped back into the boiler at T_2. When the engine exhausts into the atmosphere whose temperature is T_2 it is apparent that there is a jolt in the heat cycle, representing the difference usual between the temperature of the exhaust-steam and the temperature at which the feed-water can be usually presented to the boiler. If the feed-water is preheated by a heat-supply outside of the boiler-furnace, and other than a wasted heat, this must be allowed for.

(15) The ideal cycle assumes that the unit weight of medium is raised from T_2 to T_1 by an adiabatic compression. In actual conditions there is a gain in entropy in the gradual

heating by the fire from T_2 to T_1, and a loss because this gain is not all made at T_1.

(16) During the cushion-compression heat is transferred to the medium by the mechanical work, and some of this increase in heat condition is transferred to the metal of piston-head, cylinder-head, walls, and passages.

(17) The loss of heat represented by the equivalent of the mechanical energy consumed wastefully in the friction of the engine mechanisms, which is caused by the size and weights of the parts, packing-friction, valve-friction, and the like, which are independent of the load or work done.

(18) Any additional friction (usually small in amount) chargeable to the load of the engine in augmenting its friction when running with no load.

The above list enumerates the points in which every actual engine is likely to differ from every other actual engine, and the losses which for this reason are incapable of being included under a generalization, and are to be experimentally determined for each engine or each type of engine. For this reason they have been called *extra-thermodynamic* losses, or internal wastes. It is the object of the engineer or designer to reduce these losses as far as possible in new constructions, and in making guarantees as to performance he should be able to evaluate their proportion to the whole expenditure of energy. It will be convenient for their further study to group the principal sources of heat loss into the following headings:

I. Loss by wire-drawing.
II. " " clearance.
III. " " condensation, initial.
IV. " " " during expansion.
V. " " re-evaporation.
VI. " " incomplete expansion.
VII. " " unnecessary back-pressure, or imperfect vacuum.

The other losses are usually met by an allowance in cylinder volume to overcome them, and by that increase in cylinder volume their effect is introduced into the groups above, and need not be accounted for a second time. It is safe to say that at the end of the nineteenth century the thermal wastes are not far from 20 per cent, and the dynamic losses are less than 10 per cert.

222. Progress in Steam-engine Efficiency.—The early historic engines of Savery and Newcomen in England suffered greatly from heat-wastes, by reason of their using the working-cylinder barrel as a place within which to condense the steam after the stroke. James Watt's invention of the separate condenser in 1769 was a most important step toward reducing condensation. The following diagram, Fig. 92, has
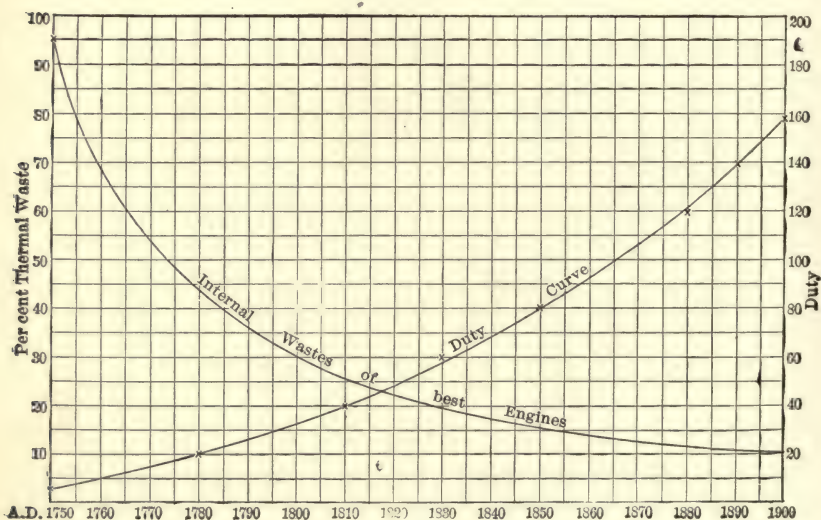
Fig. 92.

been prepared to present to the eye the improvements of the last one hundred and fifty years, showing at the left the reduction of thermal wastes by devices to be discussed presently, and at the right hand the corresponding increase in

performance per hundred pounds of pure carbon burned. This latter is given in millions of foot-pounds of work per 100 pounds of pure carbon.

The elements which seem to have had most influence in this result may be said to be:

1. The separate condenser.
2. The higher temperature and pressure limit.
3. The steam-jacket.
4. The multiple-expansion principle.
5. Better mechanical construction.
6. Higher piston speed.
7. Increasing size of units.

223. Ideal and Actual Efficiency Compared.—The elements which affect actual efficiency are numerous, and hence care is necessary in making comparisons to select fair figures. Certain of the losses, being practically fixed in amount and not dependent upon cylinder volume, will be much greater a proportion or percentage in a small engine than in a large one, and will be greater when the large engine is running below its rated power or at best effect. An observed result, with a 200-H.P. engine at 100 pounds boiler-pressure, with a back-pressure of 5 pounds above vacuum in its condenser, gave results at various ratios of expansion which are given in Fig. 93, on which the Rankine ideal water consumptions have also been plotted, and a curve for the friction and radiation losses.

The thermal units per H.P. are taken as 1000 for each pound of water.

If pressures belonging to the temperatures T_1 be made to vary, curves similar to those in Fig. 94 will result. These belong also to the simple condensing engine of average size, and larger engines will do better, but small ones not so well.

The effect of increasing the ratio of expansion by the expedient of expanding continuously through two or three cyl-
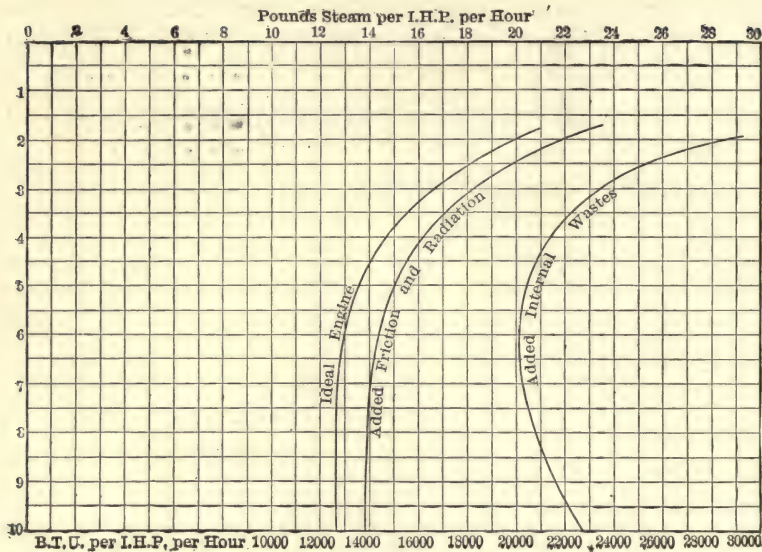
Fig. 93.
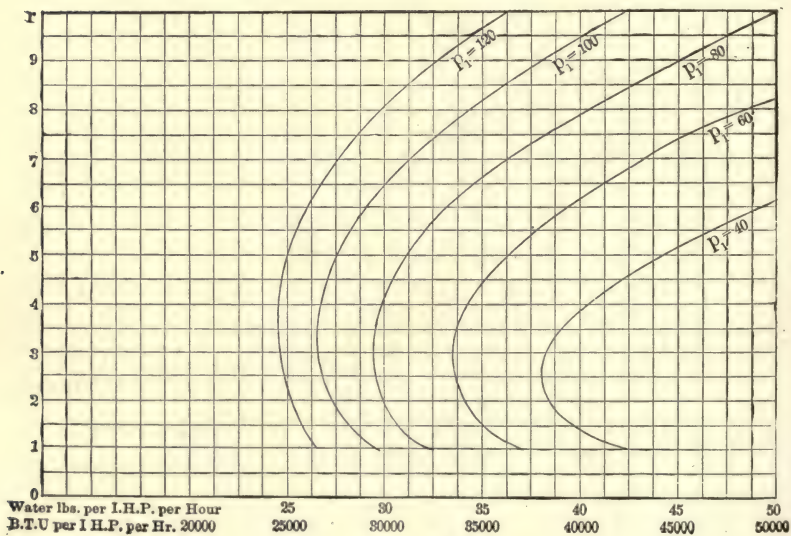
Fig. 94.

inders is made apparent by Fig. 95. The curves A are those for the smaller cylinder of a 9- and 16- and 24-inch diameter triple engine, with 36-inch stroke, working as a simple en-
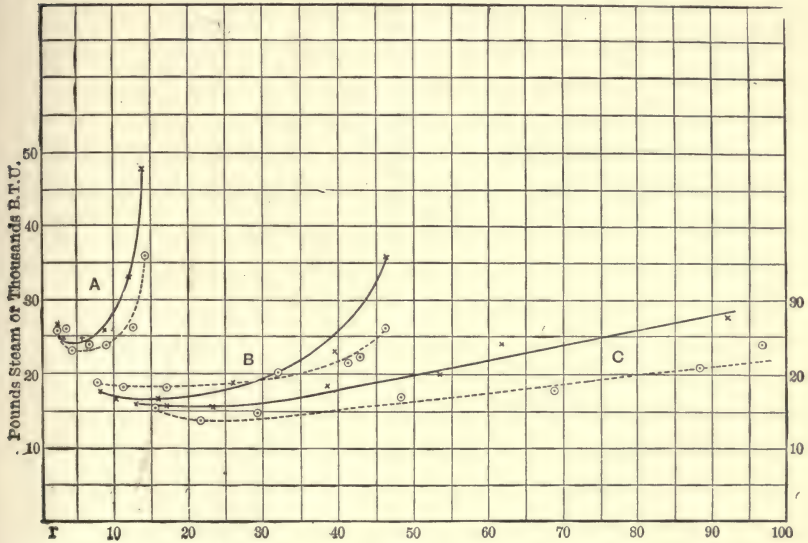
Fig. 95.

gine. The curves B are derived when the small cylinder and intermediate are used to form a compound engine; the curves C represent the triple engine, with the corresponding ratios of expansion possible. The dotted lines give unjacketed conditions; the full line shows results with cylinders jacketed.

The table on page 320 also gives water consumptions from test and experiment.

224. Methods of Reducing Internal Condensation.—It will be apparent that the compound or multiple-expansion engine should offer the advantage of diminished wastes from condensation, when it is remembered that the largest cylinder in the series is the unit cylinder, which determines the horsepower of the engine and must be present whatever system of

b	*e*	*f*	*e*	*f*	
Type of Engine.	Feed-water per Indicated Horse-power per Hour.				Per Cent Gained by Condensing.
	Non-condensing.		Condensing.		
Name.	Probable Limits.	Assumed for Comparison.	Probable Limits.	Assumed for Comparison.	
	Lbs.	Lbs.	Lbs.	Lbs.	
Simple high-speed.....	35 to 26	33	25 to 19	22	33
Simple low-speed......	32 to 24	29	24 to 18	20	31
Compound high-speed.	30 to 22	26	24 to 16	20	23
Compound low-speed..	....	*	20 to 12¾	18	25
Triple high-speed......	27 to 21	24	23 to 14	17	29
Triple low-speed......	....	....	18 to 12¾	16	....

* The terms " high-speed " and " low-speed," it is believed, refer to the number of revolutions per minute, and not to the piston-travel. Low-speed engines are Corliss engines and the like, with releasing cut-offs, and have a rotative speed usually less than 120 revolutions per minute.

expansion is used. This is because in the horse-power formula *PLAN* refers to a pressure over an area, or a *pv* result, in which the volume is always the volume at the end of the completed stroke. What has been done is therefore to add certain smaller cylinders between the boiler and the largest low-pressure cylinder, among which the temperature range is distributed, each taking a fraction of the range. Reactions between the cylinder-metal and the hot medium within it are active in proportion as the difference in their temperatures is large. By diminishing the temperature range in each, the expansion in each cylinder approaches the adiabatic law, and heat is saved.

Secondly, compounding utilizes in part at least, in the later cylinders, the steam vaporized by the metal wall reactions in the early cylinders. Some heat is thus recovered which would have been wasted in the simple engine.

Thirdly, the succession of the cylinders permits a regenerating of the quality of the steam by reheating between cylinders.

Fourthly, the compound engine favors a high value for T_1 and hence a high range in availability, and a high efficiency value or a low value for M (§ 188) when T_2 is fixed by limitation.

The compound or multiple-expansion principle has many and other advantages (see § 173). These are sufficient, however, to more than offset the losses caused by the succession of cylinders, the losses in passing from one to another, and, where fuel cost or the size of the plant will warrant it, to neutralize the increased cost of the additional cylinders and mechanism. For the specific object of reducing condensation in the steam-cylinder, the steam-jacket, and the use of superheated steam, are to be particularly discussed.

225. The Steam-jacket.—The steam-jacket was first devised and applied by James Watt, "to keep the cylinder as hot as the steam which enters it." Constructively, it is an annular space surrounding the cylinder-barrel and chambered spaces in the cylinder-heads into which steam hot from the boiler shall be kept actively circulating. Such boiler-steam shall continually put back into the metal of the working barrel the heat swept out at the exhaust from evaporation during the expansion and from contact with the relatively cool exhaust-steam. It is intended, therefore, that internal initial condensation shall be made less because the working charge of steam finds the cylinder hotter when it enters it than when such jacket is not present nor in action. Furthermore, during expansion after cut-off, the barrel-jacket will furnish the heat for any re-evaporation, or shall heat the cylinder-metal again after it has furnished the heat energy represented by such re-evaporation of water either the result of adiabatic expansion or present as remains of initial condensation. Evaporation of water mechanically entrained may also occur to cool the metal walls.

The structural difficulties which must be met in casting and using a cylinder with hollow walls (particularly when the

length is considerable) have been elsewhere considered ("Power Plants," p. 291, § 162). The unequal expansion is likely to make one wall crack; or if made in separate cylinders, the inner fitting as a liner within the jacket, the expense of fitting and the joint at the end are difficulties.

In comment on the steam-jacket it may be said:

(1) The heat which the jacket supplies to the cylinder-metal is surrendered at the cost of its own condensation. Hence the net gain from jacketing is the algebraic sum of a loss and a gain.

(2) The heat-supply to prevent initial condensation is mainly from the heads, and from the piston if this latter is also jacketed, because the barrel surface only becomes equal to the head surface when the piston has travelled a distance equal to one quarter of the cylinder diameter.

(3) After cut-off and during expansion the jacket would appear to be supplying a heat-flow to the gradually cooling steam which tends to transform the expansion curve from an adiabatic towards an isothermal, and furnish an entropy change opposed to maximum efficiency in Carnot's cycle.

(4) The hot live-steam jacket, keeping the average temperature of the cylinder higher than when the latter is not jacketed, will increase the loss by exterior radiation, unless at the same time increased precautions are taken by use of lagging and non-conductors of heat to diminish this action both at heads and at the barrel.

(5) The effective action of the steam-jacket demands that the walls of the " liner " or barrel proper should be highly conductive of heat. Transfer of heat by contact and conduction is very rapid, but cannot be truly instantaneous. Hence it would appear that jackets are more effective when their time of action on the working steam is lengthened, and when the weight of the working steam is less in proportion to the quantity of heat in units present in the jacket. The first statement is confirmed by the generally observed fact that an

engine of slow rotative speed derives more benefit from jacket-
ing than quick-acting engines; and the other statement is con-
firmed by the fact that the greater the ratio of expansion, or the
earlier the cut-off in the single cylinder or in the compound,
the greater the gain from jacketing. The time of most effect-
ive action in heating is that from cut-off on one stroke to the
beginning of admission at the next. Hence long-stroke en-
gines gain less by jacket action than short-stroke engines with
the same ratio of expansion.

(6) The gain from the jacket is proportionally less in mul-
tiple-expansion engines than in single-cylinder engines in
which high expansion is attempted. This follows because
the less the amount or tendency to condensing action the less
good the jacket can do. The divided temperature range in
the compound or multiple series diminishes the actual con-
densation, and the increased cylinder surface increases loss of
heat in the jackets themselves. Marine tests have shown a
gain from jacketing the larger and cooler low-pressure cylin-
der of a compound engine, but no gain from jacketing the
high-pressure cylinder.

(7) If the condition of high grade of expansion should hap-
pen to concur with a supply of steam initially wet, the jacket
during expansion will evaporate more water than was initially
condensed against the walls. Condensation of steam in the
jacket being a wasteful method of evaporating this water in
the working barrel, the steam-jacket may cost more than it
saves. In other words, with dry steam the jacket saves;
with wet steam, the jacket condensation may offset the gain;
when the boiler primes, the jacket is likely to be a loss.

(8) Hence if the steam is superheated, there is no occa-
sion for a steam-jacket.

**226. Conditions and Action of an Effective Steam-
jacket.**—Circulation of the hot steam from the boiler is the
prime condition of effectiveness in a steam-jacket. As the
water is condensed in the jackets it should be removed by

traps or by gravity back to the boiler.   The plan of passing the steam through the jackets of a main engine to operate the cylinder of an auxiliary engine like an independent air-pump has much to commend it.   The plan of jacketing so that the steam from the jackets enters the valve-chambers is also a good one, provided the supply to the valves is kept thoroughly dry.   Hot water in jackets is fatal to economy.   The conductivity of water for heat is very low, and a thin film of water will seriously impair the transfer of heat to the metal behind it.   In fact, this peculiarity of a water-film or dew on the inside of the cylinder often seriously disturbs the normal or anticipated action of the jacket from without.   The vertical cylinder might be expected to free itself of a water-film better than a horizontal one.   Jacketing of pistons has not been found to work well.

227. Gain from the Use of the Steam-jacket.—The gain fom the use of the steam-jacket in economy of fuel is a matter to be experimentally found for each engine and each set of conditions.   It may be a quantity varying from zero to 15 per cent, rarely reaching 20 per cent.   The expenditure of steam in the jackets is likely to be over 5 per cent in single engines and about 10 per cent in compounds, and 15 per cent or less in triples if all cylinders are jacketed.   The gain from the use of jackets results from the fact that for every pound condensed in the jackets some greater quantity is saved in the cylinders.   In the case of initially poor engines the net gain from jacketing may reach 20 or 25 per cent. With well-designed engines, such as are met in marine practice, the gain or economy is not likely to exceed 10 per cent of the total feed-water evaporated.   It belongs to the finance of the problem to decide whether the cost of the extra construction is justified by the decrease in running cost which follows from it.

The jacket results in a notable convenience in starting

engines, since the barrel and all parts of the cylinder can be warmed up in advance of the actual starting of the engine. This avoids annoyance from condensation of water, and from any seizing of fitted parts by difference of temperature.

228. Non-conducting Cylinders.—It has also been sought by certain skilful designers to mitigate the evils of internal condensation waste by making the cylinder-wall to possess such a non-diathermanous character that no reactions should take place between the steam and the metal. These objects have been aimed at either by a lining within the metallic cylinder, or by treating the metallic surface exposed to steam. The practical difficulties of a glazed or enamelled surface have arisen from the unequal coefficients of expansion of the body of the metal and the non-conducting coating, whereby the surface cracks and disintegrates. The difficulties caused by abrasion have to be overcome in the other system. These ideas, if a successful method could be found for carrying them out, would bring the actual engine up to the ideal conditions so far as internal wastes were concerned.

229. Superheating, to prevent Cylinder Condensation. —The temperature-entropy diagram in § 203 has made it apparent that it was possible to add heat to the incoming steam to such a degree that all the cooling which it must have to undergo in expansion should not be able to bring it down to the point of saturation, when it is just ready to condense to a mist on further cooling. It is obvious then that the initial condensation upon entry into the cylinder can be prevented by superheating to a less degree, and the losses thus avoided or reduced. For example, let it be assumed that such initial steam come in in a saturated state with a total heat of 1250 thermal units per pound, and that cylinder condensation under these conditions would cause a loss of 20 per cent, or that $1250 \times .20 = 250$ British thermal units disappeared into the metal walls of the cylinder by such condensation. There

must therefore be brought in by the steam, if its specific heat be called 0.480, an amount of heat represented by

$$250 = 0.480t°,$$

or the range above the working temperature of saturation for each pound becomes

$$t = \frac{250}{480} = 520° \text{ F.},$$

which is practically unattainable for reasons shortly to be treated. The same result could have been reached by the graphic process.

On the other hand, moderate superheating, of 100° to 150° F. above working pressure, is sufficient to reduce initial condensation greatly, if not to eliminate it, and when the steam is dry and the cylinder-walls are hot at the point of cut-off the evils from condensation and evaporation during expansion are mitigated. The two sets of curves shown in Figs. 96 and 97, derived from the Regnault experiments, make it plain that as the temperature increases there is a wider margin or range for a change of condition at the upper ranges than at the lower. Or in other words, the similarity of the curves shows how much more cooling has to be done at the higher levels to produce the same change which at the lower points will occur so easily.

The exceeding rapidity with which cooling takes place in the thin film of metal when acted on from within makes it also particularly rapid in responding to the heat effect of extra-hot steam coming in as a superheated gas at admission. It is for this reason that superheating has an advantage over jacketing. The jacket supplies heat not only at admission, but also during the exhaust stroke, when it is wasting it, to atmosphere or to the condenser. Superheating supplies heat only where it is required, if the initial condensation only is to be considered.

The effects of superheating the steam are:

(1) To raise T_1 in the efficiency formula, without such increase in p_1 as to give rise to practical difficulty. This increases thermal efficiency.
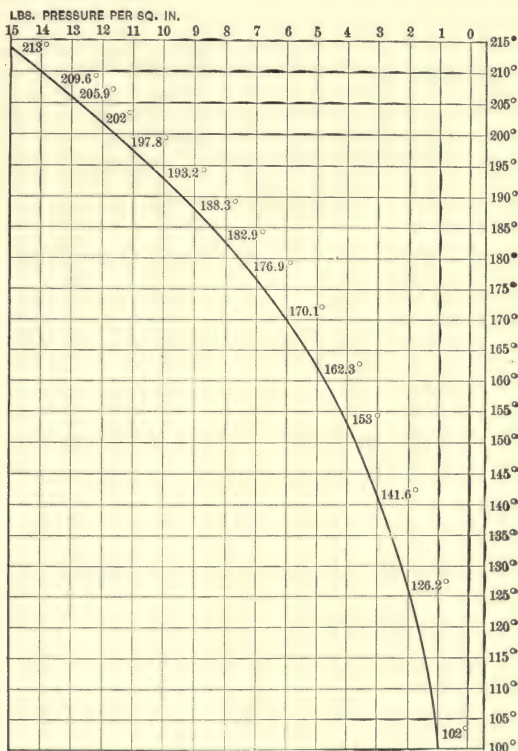
FIG. 96.

(2) To diminish the density of the steam-gas. Hence a less weight goes to an initial volume introduced into the cylinder with a given period of admission.

(3) The steam has more of the reluctance to part with its heat which is the property of a gas as distinguished from a vapor which is ready to transfer its heat to solid objects by

condensation upon them.   Saturated or wet steam condenses freely; steam-gas is difficult to cool except by intimate contact with all parts of it, as is the case with air and other gases.

230. Methods of Superheating.—There are three general principles underlying the attainment of superheat in steam.
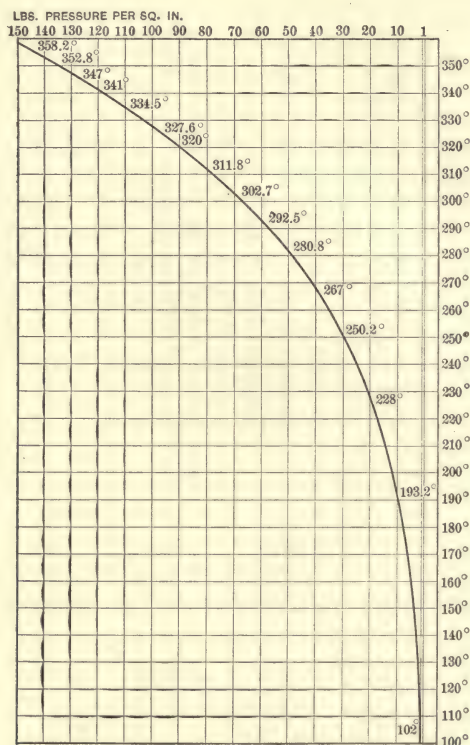
FIG. 97.

The first has already been referred to (§ 132), where two masses of steam are supposed to be separated by a throttling orifice, and flow to take place from the higher pressure into the lower pressure volume.   All the heat which was in the steam at the higher pressure is in the mass at the lower pressure, and this latter must therefore have more heat than is due

to its pressure or is superheated in consequence. That is, if the difference $T_1 - T_2$ of their total heats be one degree, then

$$(T_1 - T_2)0.480 = 1,$$

or there will be a surplus of heat represented by

$$\frac{1}{.480} = 2.08$$

degrees for each thermal unit difference of their total heats at the different pressures.

This method of superheating by wire-drawing occurs in the throttling engine, such as the locomotive, and with throttling governors on stationary engines. It is not available, however, when maximum output of energy is required.

The second method is by an admixture of highly superheated steam (usually secured by the third method) with the ordinary or saturated steam. This method of mixture has been called " adheating," or the "combined steam " process; also Wethered's system. The claimed advantage was the control of the degree of superheat by the proportions of highly superheated steam to be mixed with normal steam as the load might vary.

The third and most usual method is the direct method of heating all the steam by passing it through pipes or coils on its way from boiler to engine, such pipes being kept at high temperature by waste-heat from the furnace-gases. This result is secured in many ways:

(1) Superheating coils in the flues or at the base of the chimney (Fig. 98).

(2) Superheating in the boiler, by having a part of the heating surface above the water-line, and forcing the steam into contact with such superheating surface. Corliss and Manning boilers exhibit this method; also the common upright boiler, and the steam-chimney of the marine boiler.

(3) Superheating by surrounding the cylinder with flue-gases. This is a very old plan, but troublesome.
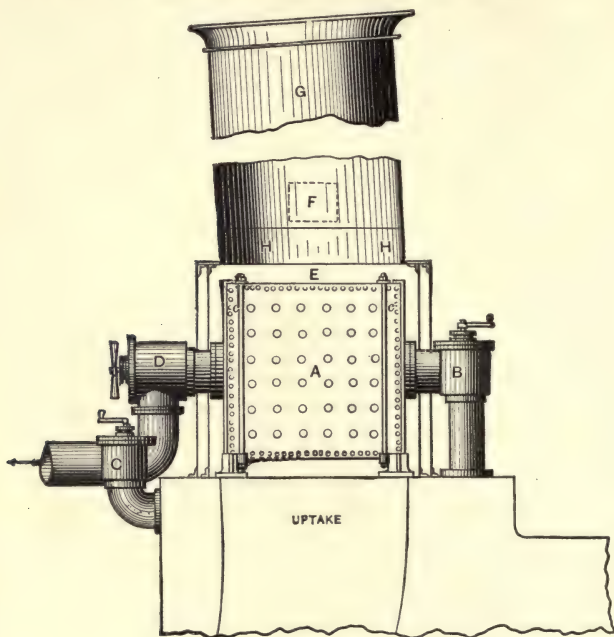
FIG. 98.

(4) In multiple-cylinder engine-practice the use of coils of steam at high or boiler pressure, in receivers through which the lower pressure steam passes on its way from cylinder to cylinder.   This is called " reheating."

231. Objections to Superheating.—European engineers have paid more attention to superheating the steam than American engineers have.   The reasons for this have been the practical difficulties in the way.   These are:

(1) The low specific heat of steam-gas causes the superheating coil to become highly heated in the furnace-gases. Hence these coils oxidize or burn out and give way.

(2) The range of temperature and expansion in superheating coils or tubes makes it difficult and costly to keep joints tight.

(3) The high temperature of the steam compels the use of metallic packings and non-oxidizable lubricants, to withstand the heat. Fibrous packings and non-mineral lubricants are impossible.

232. Gain or Economy by Superheating.—The actual gain from superheating in any case should be a matter of experimental determination, as in the case of the steam-jacket. It is furthermore complicated by the cost of renewing the direct superheater at frequent intervals, and the repairs to it. Neglecting this element, however, and speaking generally, it seems safe to say that a superheat of from 15° to 20° F. will effect an important gain in reducing wastes, and a superheat of 100° F. will practically extinguish initial condensation. With compound engines a superheat of 100° produces an average gain of 20 per cent of the fuel used with saturated steam alone. That is, the algebraic sum of the gain and the heat expended to produce the superheat will always be a positive quantity, because the return will be from twice to ten times the expenditure, taking the average of recorded tests.

233. Loss by Clearance.—There must be linear clearance between the piston and the two cylinder-heads, and a clearance volume in the passages below the values which control inlet and exit of working steam. If no compression is used or desirable, a volume of steam is taken from the boiler at each stroke and wasted, and the mean pressure is less than it would be if a smaller volume were expanding after cut-off.

The clearance volume in any actual case is found from drawings, or better by pouring in water behind the piston on its dead-centre until the clearance volume is filled. The observed weight or volume of such water gives a volume in cubic inches or cubic feet to be added to the real piston-displacement for each stroke to give the actual weight or volume fed to the cylinder per stroke.

In representing the clearance on the pv or indicator diagram, it is only necessary to prolong the diagram at its admis-

sion end by a length which shall give to the admission area the same percentage of increase in area as the clearance volume adds to the piston-displacement. (See also § 186).
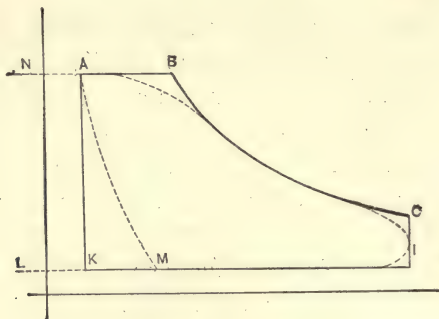
FIG. 100.

That is, if C (Fig. 100) denote the clearance volume, expressed as a fraction or percentage of the piston-displacement volume, which latter will be the product of the area A into the length of stroke L, then

$$C = fAL,$$

or

$$f = \frac{C}{AL}.$$

Therefore a length of diagram is to be added to the admission area of the indicator-card which shall be

$$fL = \frac{C}{A},$$

and the line of zero clearance drawn through a point outside the line of furthest stroke as far beyond that point as is given by the ratio $\frac{C}{A}$. In the diagram

$$LK = NA = f(AL), \quad \text{since} \quad \frac{LK}{KI} = \frac{C}{A}.$$

Neglecting clearance, with admission AB on the P.V. diagram, the apparent cut-off is

$$\frac{AB}{KI} = \frac{1}{r},$$

and the apparent ratio of expansion is

$$\frac{KI}{AB} = r.$$

But the real admission volume is NB, and the final volume LI; whence

$$\frac{1}{r_1} = \frac{NB}{LI} = \frac{AB + f}{KI + f} = \frac{\frac{1}{r} + f}{1 + f},$$

and

$$r_1 = \frac{LI}{NB} = \frac{1 + f}{\frac{1}{r} + f} = \frac{r + fr}{1 + fr},$$

so that the increase of expenditure of fluid and hence of heat becomes

$$\frac{NB}{AB} = 1 + fr,$$

while the absolute mean pressure is less than it would be if the clearance volume were not also filled with expanding steam in the proportion

$$p'_m = p_m - f(p_1 - p_m).$$

If the values of the clearance volume are not known or conveniently measurable, the line of zero volume and no clearance can be drawn with close approximation from the actual card, upon the assumption that for a short distance the compression line departs so little from an equilateral hy-

perbola that it may be called one.   Then if two symmetrical points are chosen on the curve (Fig. 101) and a line drawn through them to the line of zero pressures, which it will cut
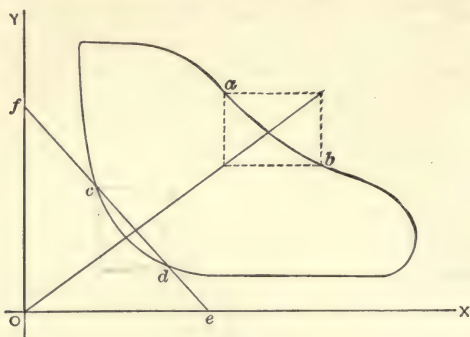
FIG. 101.

at some point e, and prolonged also beyond c, it will cut the line of zero volumes as far beyond c as the point e is beyond d.   Making $cf = de$, the line OY through f is the line of zero volumes, or the line of clearance zero.   This comes because, by similar right-angled triangles, $pv = p_1v_1$, only when O is determined in this way.   A determination of the location of the point O by two points on the expansion curve is less accurate because the curve may diverge from the equilateral hyperbola, and any errors in such diagonal line are multiplied in locating the point O.

234. Probable Amounts of Clearance.—Small engines may have the linear clearance as low as one eighth of an inch; larger engines may have as much as one-half inch.   The longer the stroke relatively to the diameter, the less the percentage of clearance represented by a linear unit of clearance.   Short-stroke high-rotative-speed engines therefore empty the greatest clearance volumes.

Corliss valve-gearing causes the least volume between valves and cylinder-bore, and the same is true for those designs which have the valves in the cylinder-heads.   Valves of

the poppet type and piston-valves compel a considerable clearance volume and attendant loss. A value for f as low as $2\frac{1}{2}$ per cent is as low as is usually obtained; it ought not to exceed 7 per cent; if it goes above 10 per cent it is excessive.

235. Clearance Losses Diminished by Compression.— The effect of a preclosure of the exhaust-valve before the end of the exhaust-stroke is to entrap steam between the piston and the cylinder-head, and to raise its pressure by compressing it into the clearance volume, which it may fill completely. The work of such compression is a negative work so far as the useful work of the engine is concerned, and is done by the fly-wheel at the expense of energy stored in it. It is mechanically advantageous, however, as furnishing a gradually increasing cushion effect against the reciprocating parts to arrest them, and to put them under the strain of tension or compression to which the next working stroke is to subject them. The temperature-entropy diagram shows the point at which this compression should begin if adiabatic compression of the entrapped steam-vapor is to raise the pressure to that corresponding to T_1. (See § 200.)

236. Calculation of Mean Effective Pressure when Clearance and Compression are Considered.—The ac-

FIG. 102.

cepted method for calculating mean effective pressure when the diagram of the indicator shows both clearance and com-

pression is an extension of principles already laid down.   If L in Fig. 102 is the length of the stroke, l the length of the admission line, x the period of the exhaust-stroke after compression begins, c the clearance length proportional to the clearance volume, while the capital letters A, B, C, and D represent the areas on which they are placed, and the pressures are respectively p_1 at admission, p_b during exhaust, and p_c at the end of compression, we shall have the total area of the enclosing figure from §§ 164–167:

$$\text{Area of } ABCD = p_1(l + c)\left(1 + \text{hyp. log} \frac{L + c}{l + c}\right).$$

But by similar reasoning the area of the parts B, C, and D will be given by the equations

$$B = p_b(L - x);$$

$$C = p_c c\left(1 + \text{hyp. log} \frac{x + c}{c}\right)$$

$$= p_b(x + c)\left(1 + \text{hyp. log} \frac{x + c}{c}\right);$$

$$D = (p_1 - p_c)c = p_1 c - p_b(x + c).$$

Hence the area of the net-work diagram A will be

$$\text{Area of } A = ABCD - (B + C + D)$$

$$= p_1(l + c)\left(1 + \text{hyp. log} \frac{L + c}{l + c}\right)$$

$$- \left[p_b(L - x) + p_b(x + c)\left(1 + \text{hyp. log} \frac{x + c}{\cdot}\right) + p_1 c - p_b(x + c)\right]$$

$$= p_1(l + c)\left(1 + \text{hyp. log} \frac{L + c}{l + c}\right)$$

$$- p_b\left[(L - x) + (x + c) \text{ hyp. log} \frac{x + c}{c}\right] - p_1 c.$$

Hence, since the mean effective pressure will be the result of dividing the area of the work-diagram by its length, we have

$$\text{Mean effective pressure} = \frac{\text{area of } A}{L}.$$

237. Friction in Steam-pipes.—When the volume of steam required by the engine is known per unit of time, experience shows that loss of pressure and temperature from friction or wire-drawing will be inappreciable if the cross-section of pipes or passages is so made that the linear velocity of the steam shall not exceed 100 feet per second. Length is not without effect, but for short distances, and where engine and boiler are close together, the velocity may be increased.

238. Loss of Pressure and Temperature from Cooling in Pipes.—When steam is moving in pipes which are adequately clothed with non-conducting coverings there is no doubt a loss by eddies and by a higher velocity at the beginning than at the end, due to the long travel. Experiments, however, on any considerable or adequate scale are lacking to enable this difference to be more than guessed at, under the wide variation of condition as to exposure and effectiveness of covering. Designers usually allow for a loss of 5 per cent of pressure in long runs of over 250 feet, and allow variations from this allowance by climate and season and deterioration of the insulation to be met by changing the pressure at the source of heat.

239. Efficiencies Experimentally Determined in Terms of Thermal Units.—The calculation of § 218 showed that

$$\frac{33000}{778} = 42.164$$

thermal units per minute per horse-power would be required by an ideal engine.

If an actual engine be tested and found to consume n

pounds of water per horse-power per hour, it will be obvious that the thermal units consumed by that engine per horse-power will be the product of the weight n into the specific heat unity into the range of temperature used by the engine. This range will be the difference between the total heat of the steam at the temperature used and the temperature of the feed-water as supplied to the boiler. Hence

$$\text{Efficiency} = \frac{42.4164 \times 60}{n \times (\lambda - t)},$$

when λ is the total heat at T_1, and t is the temperature of the feed-water. Usually the boiler is combined with the engine, and the efficiency is thus taken together. In the case, however, where it is desirable to separate them, the calorific power of the fuel being known, and the total heat above feed-water temperature being given, it will be obvious that the product (total heat — feed-water heat) × (pounds of water so heated) should be equal in theory to that calorific power. The efficiency of the boiler should be the ratio:

$$\text{Efficiency} = \frac{\text{Actual pounds evaporated per pound of fuel}}{\text{Theoretical evaporation per pound of that fuel}}.$$

Then the efficiency with the theoretical cycle having been calculated between the limits T_1 and T_2, or the theoretical water consumption, the actual water consumption is compared with the theoretical, the latter being taken as 100 per cent; then

$$\text{Efficiency of engine} = \frac{\text{Actual water per H. P. per hour}}{\text{Theoretical water per H. P. her hour}}.$$

The combined efficiency is the product of the two efficiencies in percentage.

This method is fairer than the comparison with unity of perfection in the Carnot cycle. The difficulty with the steam-engine is that the fuel-temperature of 2000° F. in the furnace

gives as yet only a 400° temperature of the heat medium for physical reasons. While the limits imposed by temperature are such as yet as to keep theoretical limits of efficiency far beyond our present practice, and induce earnest research either after media which shall not be subject to these limitations, or to extend the limits, yet on the other hand it is unfortunate not to be able to appreciate how excellent our heat-engines are when the perfection realizable within these limits is taken into the calculation.

CHAPTER XVII.

THERMAL ANALYSIS OF HEAT-ENGINES.

240. Introductory.—It has been repeatedly emphasized in previous chapters that the character of the reactions causing loss of heat and efficiency, and the action of the appliances devised to mitigate these losses, were matters demanding experimental investigation for each particular engine. Skilful designers however must have general principles to use, and their skill will consist in the wise application of these to the problem then in hand. But the ultimate criterion even in duplicate engines must be the actual test and the analysis of the results.

The testing of engines and the interpretation of the data of such tests have long been matters interesting the foremost grade of practitioners, and form a field too wide to be entered on here except in a summary way. The use of the indicator and the deductions from its diagram with respect to distribution, value for mean effective pressure, and horse-power must be studied elsewhere. This chapter will discuss only the deduction of water per horse-power, the Hirn analysis, and the temperature-entropy diagram as giving the distribution of the heat energy.

241. Pounds of Heat Medium per Horse-power Calculated Theoretically from an Indicator-diagram. — The steam used in a steam-engine weighs the same as the water furnished to the boiler (less wastes) when observed over a long enough interval. The heat delivered to the engine will be proportional to the weight of steam which it consumes.

Hence the most satisfactory test is to weigh the water used by the engine by catching it in a surface condenser. Where this cannot be done, and often also where it is possible, it is desirable to determine, from the indicator-card of data on the pv plane, what weight of water the diagram corresponds to. In Fig. 103, let L be the length of the stroke in feet; A the area
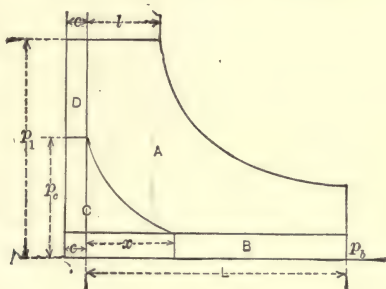
Fig. 103.

of the piston in square inches, so that $\dfrac{A}{144}$ is the area in square feet; $N =$ number of strokes per minute; l the percentage of the stroke completed at the point of cut-off, if the water rate is to be computed for that point, or at the release, if the rate is to be there computed; c the percentage which clearance volume bears to piston-displacement, and hence the same relation to the stroke length; w the weight per cubic foot of steam at the pressure at which the water rate is to be calculated, and w' the weight belonging to the pressure at the end of any compression that there may be. Then

$$\text{Cubic feet per stroke} = L\left(\frac{l + c}{100}\right)\frac{A}{144}.$$

$$\text{The clearance volume will be} = \frac{LcA}{144 \times 100}.$$

The weight of steam in pounds per stroke will be the cubic feet times w, or

$$W_1 = L\left(\frac{l+c}{100}\right)\frac{A}{144}w,$$

and the weight in the clearance volume w_2 will be

$$w_2 = \frac{LAcw'}{14400}.$$

The total weight per stroke will be the difference between W_1 and w_2, or

$$W_1 - w_2 = L\left(\frac{lAw + cAw}{14400}\right) - \frac{LAcw'}{14400}$$

$$= \frac{LA}{14400}[(l+c)w - (cw')].$$

This becomes weight of water per hour by multiplying by $60N$, or

$$W = \frac{60LAN}{14400}[(l+c)w - (cw')].$$

In this result all data are on the indicator-diagram, or are from tables and observations. To reduce this to weight of water per horse-power per hour, both members are to be divided by the equality

$$\text{H.P.} = \frac{PLAN}{33000},$$

in which P is the mean effective pressure from the diagram. Hence

$$\frac{W}{\text{H.P.}} = \frac{\dfrac{60LAN}{14400}[(l+c)w - (cw')]}{\dfrac{PLAN}{33000}} = \frac{137.50}{P}[(l+c)w - (cw')].$$

For compound or multiple engines P will be the mean pressure reduced to the low-pressure cylinder volume, assuming all work to be done in that cylinder.

This equation repeats the statement of § 236. If the pressure in the clearance is carried by compression so that $w' = w$, then

$$\frac{W}{\text{H.P.}} = \frac{137.50}{P}(lw).$$

If, however, there is no compression, and $w' = $ zero, then the consumption per horse-power is

$$\frac{137.50}{P}(l + c)w.$$

The difference between the water supplied to the engine and the value for W from the card is known as " water not accounted for by the indicator " which has been expended in the initial condensations, leakages, etc., which constitute the losses hitherto discussed.

The above values for w refer only to points chosen between cut-off and release. The results at these two terminal points are likely to differ. The amount of their difference is a rough gauge of the amount of re-evaporation in the cylinder. The indicator will imply the greatest weight at release for this reason.

242. Hirn's Analysis.—The thermal analysis most in use in America and Europe was first proposed and elaborated by the great Alsatian engineer G. A. Hirn, and applied by his distinguished pupil and colaborer Octave Hallauer. In the form most used it bears the impress of later study by Prof. V. Dwelshauvers-Déry of Liège, Belgium. The first step in applying it is to select a representative indicator-card, whose curves on the pv surface shall give the normal performance of the engine over a considerable time, and representing practically constant conditions. A weight of steam M in pounds

is supplied to the cylinder per stroke (or per 100 strokes if desired) occupying the volume up to the point of average cut-off by diagram, which will be denoted by V_1. The closure of the exhaust-valve on the previous stroke has entrapped a volume V_0 or a weight M_0 of steam and water mixture in the clearance volume (V_0). The quantity M may be called the cylinder-feed; the quantity M_0 may be called cushion-steam and is found by selecting on the average card the earliest point at which the valve is known to be closed on the exhaust-stroke, calculating the volume from this abscissa, and adding the clearance volume. The weight for this pressure and this volume is found from tables. It is assumed that the steam is dry and saturated at the compression-point.

The weight M is of course most satisfactorily found by measurement directly from a surface condenser where this is possible; if not convenient, the feed to the boiler should be the same as the feed to the cylinder if no other apparatus is supplied by the boiler, such as pumps, jackets, injectors, leakage, and the like. To make the case general, it should be assumed that the steam is wet, or that a percentage x is vaporized, while $1 - x$ remains as water. Hence the volume of one pound of the mixture, if u represents the increase in volume of the water when it becomes steam and σ is the volume of the liquid water,

$$v = xu + \sigma,$$

and for M pounds this will be

$$Mv = V = M(xu + \sigma).$$

If, then, the subscript letters represent the various points at which volumes are noted on Fig. 104, and the corresponding states of the mixture, we have

$$V_0 = M_0(x_0 u_0 + \sigma) \qquad \text{for admission;}$$
$$V_0 + V_1 = (M + M_0)(x_1 u_1 + \sigma) \quad \text{`` cut-off;}$$
$$V_0 + V_2 = (M + M_0)(x_2 u_2 + \sigma) \quad \text{`` release;}$$
$$V_0 + V_3 = M_0(x_3 u_3 + \sigma) \qquad \text{`` compression.}$$

Assuming evaporation complete at compression makes this last equation present x_s as unity, and therefore

$$M_0 = \frac{V_0 + V_s}{u_s + \sigma}.$$

But $u_s + o$ will be the volume of one pound of completely evaporated steam at the point of compression V_s, and this volume will be the reciprocal of the weight w' of one cubic foot at that pressure from tabular values. Hence

$$M_0 = (V_0 + V_s)w'.$$

This value for M_0 can be inserted in the equations above, and the values for x_0, x_1, and x_2 calculated for admission, cut-off, and release, respectively.

In the second place, the heat brought into the cylinder by
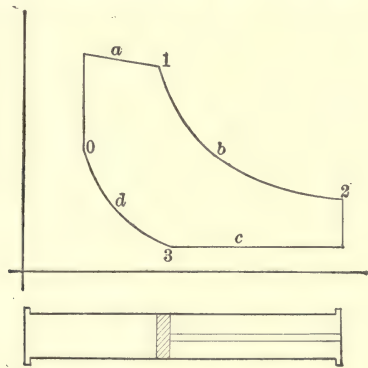
Fɪɢ. 104.

M pounds of steam will be

$$M(q + xr) = Q,$$

when q is the heat of the liquid and r is the latent heat of vaporization for the percentage x which has been vaporized.

If the steam is superheated, then Q becomes

$$Q_s = M[\lambda + .480(t_s - t)]$$

if t, is the temperature of the superheat and .480 is the specific heat of steam at constant pressure.

But it is necessary to separate the entire heat energy into those fractions which correspond to outer or external work, and those which represent intrinsic heat energy, which can be otherwise disposed of than in doing such external work. If the heat equivalent be called H, then

$$H_0 = M_0(q_0 + x_0\rho_0) \qquad \text{for admission;}$$
$$H_1 = (M_0 + M)(q_1 + x_1\rho_1) \quad \text{`` cut-off;}$$
$$H_2 = (M_0 + M)(q_2 + x_2\rho_2) \quad \text{`` release;}$$
$$H_3 = (M_0 + M)(q_3 + x_3\rho_3) \quad \text{`` compression.}$$

The symbol ρ replaces r because the factor must contain only the heat equivalent for the *internal* work of vaporization of one pound, and not that corresponding to both internal and external work of vaporization.

In the third place, it becomes apparent that when the steam entered, bringing Q units of heat, it found already the clearance volume filled with a steam whose energy was H_0. At the end of admission when cut-off takes place, an external work in foot-pounds W has been done, whose heat equivalent is $\dfrac{W}{778}$, or AW, and there remains an intrinsic energy H_1 in addition to any transfer to the cylinder-walls of heat which has disappeared in initial condensation or otherwise. Hence

$$Q + H_0 = AW_a + H_1 + Q_a,$$

or

$$Q_a = Q + H_0 - H_1 - AW_a$$

if Q_a denote such lost heat during admission.

During the expansion period external work W_b is done, and at release an intrinsic energy H_2 remains. Whether H_2 or H_1 will be numerically the larger must depend on whether the walls by jacket or otherwise supply heat to the working-

cylinder-feed steam or are withdrawing it. In any case, calling the transfer Q_b and assuming that it denotes an absorption, the intrinsic energy at cut-off must balance the work, the loss from condensation, and leave the remainder H_2 of energy to be present at release. Hence

$$H_1 = H_2 + AW_b + Q_b,$$

or

$$Q_b = H_1 - H_2 - AW_b.$$

During the third period, or exhaust, the engine is pumping out the exhaust steam, or is doing a negative work W_c, and at the end of exhaust, or when compression begins, there must remain an intrinsic energy represented by H_3. If the heat of the liquid water resulting from the condensation in a condensing engine be represented by q_4, which is the hot-well temperature with a surface condenser, then a quantity of heat represented by Mq_4 is carried into the condenser. If G pounds of condensing water are used to effect this condensation, and delivered at injection temperature q_i are raised to the outflow temperature q_k, then an amount of heat disappears in this process of cooling which will be represented by $G(q_k - q_i)$. If a jet condenser is used, t_4 of hot-well and condenser outflow t_k will be the same. The loss to the walls being denoted by Q_c, the heat energy H_2 must balance:

$$H_2 = H_3 + Mq_4 + G(q_k - q_i) + Q_c - AW_c,$$

or

$$Q_c = H_2 - H_3 - Mq_4 - G(q_k - q_i) + AW_c.$$

If the release leads the ends of the stroke, some of the expulsion work will be done by the working fluid. If this happens, W_c will be the difference between the lengths of these two parts of the stroke in which the exhaust is done by the driving steam of the stroke, and the length of the expulsion done by the steam of the following stroke.

During the fourth or compression period a work W_d is done by the engine. Any losses Q_d must leave the energy represented by H_0 at the end of the compression, so that

$$H_3 = H_0 - A W_d + Q_d,$$

or

$$Q_d = H_3 - H_0 + A W_d.$$

Finally, there is whatever expenditure of heat is represented by the jackets of cylinders, receivers, and the like. Let m represent the weight of water collected per stroke (or per 100 strokes), with x' its percentage of dryness, r' its heat of vaporization, and q' its heat of liquid, while q'' is the heat of the liquid withdrawn as water from the jackets; then if Q_j be the heat so supplied to the jackets,

$$Q_j = m(x'r' + q' - q'').$$

It is obvious on the other hand that the quantity Q supplied to the engine must be great enough to do the following:

(*a*) do the external work W, which will be

$$W = W_a + W_b - W_c - W_d,$$

or the work corresponding to the entire net area of the indicator-card;

(*b*) supply the heat carried into the condenser and away by the injection

$$M q_4 + G(q_k - q_i);$$

(*c*) meet all the losses by radiation, condensation, or otherwise; the summation of all the losses

$$Q_e = Q_a + Q_b + Q_c + Q_d.$$

To help out the quantity Q, the jacket heat Q_j must be supplied in addition, and it may be treated as if it were also supplied per stroke as an addition to the working fluid. Hence

$$Q + Q_j = AW + Mq_* + G(q_k - q_i) + Q_e,$$

from which it follows that, by addition of equations,

$$Q_e = Q_a + Q_b + Q_c + Q_d + Q_j,$$

and also

$$= Q + Q_j - Mq_* - G(q_k - q_i) - AW,$$

all of which latter quantities are capable of experimental observation in a properly arranged engine-test, and should check with the summation of the losses in each part of the cycle.

243. Application of Hirn's Analysis.—If a thermal analysis is to be applied to an engine-test, the foregoing equations have shown the quantities which are to be observed and recorded. For the determination of $Q = M(xr + q)$ the weight of cylinder-feed per stroke observed from the condenser is to be multiplied by the percentage of dry steam present in the cylinder-feed, and this must be determined by a calorimeter so located as to give the indication of quality which is prevalent in the cylinder. This is not always easy, and offers scope both for skill and for care.

The data concerning the cylinder must involve not only length and diameter, but also the points of cut-off, release, and compression from both head end and crank end, the clearance volumes at both ends, and the piston-displacements at both ends. These latter quantities are rarely or never the same for both ends, because the connecting-rod introduces an irregularity of path of the piston by its angular motion, and the piston-rod fills a part of the clearance volume at one end and not at the other. It is usual to take the half-sum of the volumes in working out numerical values.

Then from the average diagram the absolute pressures are ascertained by measurement, adding to them the barometer-reading on the day of the test so as to locate the atmospheric line in its proper place above a true vacuum. These pressures being observed for cut-off, release, compression, and for the point on the compression at which the inlet-valve opens for admission on both crank end and head end, it is convenient to tabulate the corresponding values for q, r, ρ, and u derived from tables; and similarly to work out the value of $\frac{W}{778}$ for each section of the card from the mean pressure prevalent when W_a, W_b, W_c, and W_d respectively is being performed. The mean pressure into the area, both in square-inch units, multiplied by the length in feet through which that effort is exerted, gives a value for W in foot-pounds, which is reduced to the equivalent in heat-units by dividing by 778, since $A = \frac{1}{J}$. The volumes in cubic feet are then calculated for V_0, V_1, V_2, and V_3 for both head and crank ends.

The longer the duration of the test, the less the chances for error, and the more insignificant its percentage. This is particularly true of the weight of steam per stroke:

$$M = \frac{\text{Total weight of steam used in the test}}{2 \times \text{number of revolutions during that period}},$$

and also for the weight of condensing water per stroke, which will be

$$G = \frac{\text{Total weight of condensing water during period of test}}{2 \times \text{number of revolutions during that period}}.$$

The temperatures of the condensed steam in the hot-well, and of the injection before and after use in the condenser,

have to be observed with sufficient frequency to represent average conditions.

These data having been prepared, M and M_0 are calculated; the values for x, H, and Q with their several subscript values are worked out from tables.

The work per stroke having been found in foot-pounds from the pv indicator-card and reduced to heat-units for the foregoing calculations by dividing by 778, the horse-power will be found by multiplying the heat-unit work by the factor 778 and dividing by the foot-pounds in one horse-power for the same period. If, as is usual, the work in heat-units *per hour* is the result of the calculation, then

$$\text{H.P.} = \frac{778 \times W \times 2Rpm \times 60}{60 \times 33000},$$

when W is the heat-unit work for one stroke. Then it is obvious that the steam per horse-power is the quotient resulting from dividing the total weight of steam used by the number of horse-power developed during that same period.

The quantity Q is, however, the principal object sought, and when found for one revolution can be expressed per minute or per hour or per horse-power as desired.

The table on page 352 will show an illustrative analysis involving the foregoing details.

244. Limitations of Hirn's Analysis.—If the engine is a non-condensing one, the quantities depending on the condenser will vanish. But since it is true that

$$Q_c = Q + Q_j - Mq_4 - G(Q_k - Q_i) - AW,$$

the equation of the previous paragraph for Q_c may be written

$$Q_c = H_2 - H_3 - Q + Q_j - Q_4 + A(W + W_4)$$

by substitution.

THERMAL ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS.

Quantities.	Symbols.	Formulæ.
1. Steam entering working cylinder, pounds.	M	. . . .
2. Steam at admission, pounds	M_0	$100 \, (V_c + V_0) \div v_0$
3. Steam used by calorimeter, pounds		. . . .
4. Steam, total, pounds	$M + M_0$	. . . .
5. Heat of condensed steam	K'	$M q_g$
6. Condensed water, pounds	G	. . . .
7. Heat given to condensing water	K	$G \, (q_k - q_i)$
8. Heat supplied to engine	Q	$M \, (xr + q)$
9. Sensible heat at admission	H_0	$M_0 q_0$
10. Internal heat at admission	H_0'	$100 \dfrac{V_c + V_0}{v_0} \rho_0$
11. Sensible heat at cut-off	H_1	$(M + M_0) q_1$
12. Internal heat at cut-off	H_1'	$100 \dfrac{V_c + V_1}{v_1} \rho_1$
13. Sensible heat at release	H_2	$(M + M_0) \, q_2$
14. Internal heat at release	H_2'	$100 \dfrac{V_c + V_2}{v_2} \rho_2$
15. Sensible heat, beginning of compression.	H_3	$M_0 q_3$
16. Internal heat, beginning of compression.	H_3'	$100 \dfrac{V_c + V_3}{v_3} \rho_3$
17. Cylinder loss during admission	Q_a	$Q + H_0 + H_0' - H_1 - H_1' - A W_a$
18. Cylinder loss during expansion	Q_b	$H_1 + H_1' - H_2 - H_2' - A W_b$
19. Cylinder loss during exhaust	Q_c	$H_2 + H_2' - H_3 - H_3' - K - K' - A W_c$
20. Cylinder loss during compression	Q_d	$H_3 + H_3' - H_0 - H_0' - A W_d$
21. Heat discharged, and work	B	$K + K' + A W$
22. Jacket	D	$Q - B$
23. Jacket	D'	$Q_a + Q_b + Q_c + Q_d$
24. Quality of steam entering	x	per calorimeter . . . . per cent.
25. Quality of steam at cut-off	x_1	$100 \dfrac{V_c V_1}{(M + M_0) v_1}$ . . . . "
26. Quality of steam at release	x_2	$100 \dfrac{V_c + V_2}{(M + M_0) v_2}$ . . . . . . , ... "
27. Quality of steam at compression . . . . , ...		$100 \dfrac{V + V_3}{M_0 v_3}$ . . . .
28. Quality of steam at admission	x_0	per calorimeter . . . . "
29. Quality of steam in exhaust	x_5	$\left(\dfrac{K + K'}{M - M_x} - q_5 \right) \div r_5 \ldots$ "
30. Heat lost, admission	a	$Q_a \div Q$ . . . . "
31. Heat restored, expansion	b	$Q_b \div Q$ . . . . "
32. Heat rejected, exhaust . . . . , ...	c	$Q_c \div Q$ . . . . "
33. Heat lost, compression	d	$Q_d \div Q$ . . . . "
34. Heat utilized, work	w	$\dfrac{W}{778} \div Q$ . . . . "
35. Heat lost, radiation	R	Radiation $\div Q$ . . . . "
36. Ratio, radiation to work	. . . .	$R \div w$
37. Ratio, cylinder condensation to work	. . . .	$a \div w$
38. Thermodynamic efficiency	E	$(t - t_3) \div (460 + t)$ . . . . per cent.
39. Actual efficiency	E_1	$A W \div Q$ . . . . "
40. Efficiency compared with ideal	E'	$E_1 \div E$ . . . . "

Special symbols, V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A = \dfrac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

The usual custom is to assume that the steam at the end of compression, or just when the inlet opens, is quite dry and saturated, or that x_0 is unity. It has been already said that the condition of the steam corresponding to x_s at the beginning of compression or the end of exhaust was considered to be that of dry steam; hence while the presence of any considerable quantity of water in the clearance volume is unlikely, it is not entirely justifiable to assume that the steam is dry. The difficulty of ascertaining the truth of this fundamental assumption has made many feel a dissatisfaction with the calculated results, and furthermore it will be apparent that the errors of observation and computation in the successive equations for Q are cumulative in their effect upon the final value for Q_e, and may make a large percentage of its value.

245. Thermal Analysis by Temperature-entropy Diagram.—The methods given for a transfer from the pv diagram to a T.E. diagram in § 205 require to be extended when clearance and compression volumes are to be considered. But the inspection of the T.E. diagram resulting from such a transfer will give the thermal analysis more clearly than the analytic method by Hirn and Dwelshauvers, although of course when correctly done the results in both should agree.

The method followed here was first advanced by Boulvin, and has been further elaborated by Prof. Reeve.

The steps for the transfer will involve much of the same procedure as above:

(1) The drawing or selection of an average and representtive indicator-card on the pv plane.

(2) The indicated horse-power is to be worked out, and the total feed-water supplied to the cylinder during the test. From this is derived—

(3) The water rate per horse-power per hour. The rate per stroke fed to the cylinder will be the total water consumption divided by twice the number of revolutions per hour.

(4) What is needed, however, is the reciprocal of (3) or

the number of strokes needed to make up a pound of the cyl-
inder feed.   In large engines this may be a fraction less than
unity; in small engines it will be a whole number, and per-
haps a large one.

(5) Find the weight of cushion-steam as in Hirn's analy-
sis.   Find first the clearance volume, add the volume given
by the indicator-card when the exhaust-valves close, and mul-
tiply this by the volume resulting from the calculation in (4).
This volume at the compression pressure can be reduced to
weight from steam-tables.   With multiple-cylinder engines
this should be done separately for each cylinder.

Checking these with the methods followed in § 243, it will
appear that (3) corresponds to M and that (5) corresponds to
M_0.   The next steps will be the proper graphical plotting of
the pv card, that the proper entropy values may be meas-
ured from it.   The object sought is to pass from the piston-
displacements given by the indicator to the volumes of heat
medium supplied to the cylinder, the temperature of such
volume being that in each case which belongs to that pres-
sure.

(6) Construct a diagram (Fig. 105) which shall represent
by the curve MN the saturation curve for one pound of
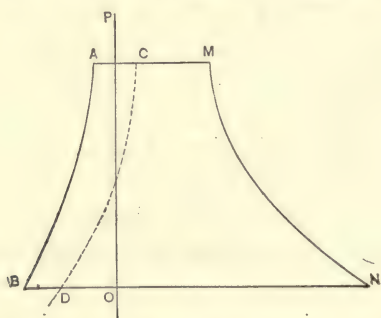
FIG. 105.

steam measured for complete vaporization from a zero line
OP, and having the height of the point M above the line ON

correspond to the absolute pressure in the boiler above the
line of perfect vacuum on the same scale as the indicator-card
or proportional to it, so that N would correspond to complete
expansion, or to the pressure corresponding to the limit value
of T_2 as may be preferred. The vertical OP would be the
point to measure the cylinder-feed volumes if there were no
clearance and no cushion-steam in such volume or in the re-
ceivers undergoing expansion and compression at each stroke,
and therefore affecting the actual volume of cylinder-feed.
Since this cushioning weight is varying in volume as the pres-
sure varies in the cylinder, the line to be drawn to the left of
OP to represent this increase in actual volume due to clear-
ance will not be a straight line, but will be an adiabatic for
the clearance volume V_0 between the limits of the back pres-
sure and the admission pressure. The methods for drawing
adiabatic curves have been given elsewhere (§§ 123 and 125).
A curve AB results in Fig. 105 which is as far to the left of
the axis OP as the varying volume of the expanding cushion-
steam adds to the volume of the cylinder-feed at any pres-
sure. The horizontal lengths between the curves AB and MN
at any pressure give the total volume of steam in the cylin-
der at that pressure, which will be the sum of the volumes
of cushion-steam and cylinder-feed steam, assuming vaporiza-
tion complete. Hence if the cushion-steam does not show
the volume to be expected, this will mean that some of it
has been condensed to meet demands for heat from the
working volume, for which heat something should give
account.

(7) If, then, horizontal lines intercepted between the
curves AB and MN measure the sum of the cushion-steam in
clearances and the cylinder-feed, a curve must be capable of
being drawn between AB and MN which shall intercept the
volume of the cushion-steam from the curve AB at each
pressure, and which shall establish the zero line of piston-
displacements, which are the units of the indicator-diagram.

This is done by measuring from AB to the right at a suffi-cient number of horizontal distances a length which shall be the product of the clearance volume into the number of strokes per pound of cylinder-feed (4). The dotted curve CD results in Fig. 105, and would appear to be the axis curve from which horizontal lines of the indicator-diagram were to be measured for each pressure, if the indicator-card had given pressures corresponding to volumes of cylinder-feed steam instead of volumes of piston-displacement only.

(8) Hence if from CD as an axis of volumes the indicator-card from the actual engine be laid down, taking vertical dis-tances as measured from the line of perfect vacuum, and lay-ing off the horizontal distances from the curve CD at the same height, a diagram of cylinder-feed volumes *indeca* results (Fig. 106). It will usually be within the saturation curve MN, be-

Fig. 106,

cause the volumes are made less than the theoretical by the condensations which measure the heat losses. The only case where the actual could pass beyond the saturation curve would be where superheating was practised.

(9) The diagram of volumes for the cylinder-feed or M can now be transferred to the theoretical temperature-entropy diagram by the principle laid down in § 205, that horizontal lines on the pv theoretical diagram and the T.E. theoret-ical diagram are divided proportionally by the points of the

actual diagram. The height of the line AB above the horizontal axis of entropy measures the temperature corresponding to the boiler-pressure; the point C is distant from the temperature axis a distance which is the entropy counted from an assumed zero, and the line CD is the increase in the entropy during complete vaporization at the temperature T_1. The line FG belongs to T_2, and its length is the measure of the decrease in entropy belonging to the feed-water temperature. The line OC is the logarithmic curve representing the gradually heating cylinder-feed, with temperature and entropy increasing together. The plotting of the diagram from Fig. 106 may cause it to appear on the T.E. diagram somewhat like Fig. 107.
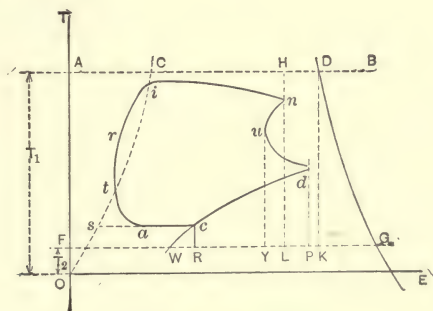
FIG. 107.

246. Losses Revealed by the Temperature-entropy Diagram.—(1) The line *in*, representing the period of admission in the actual engine, will not be as long as CD if the steam carries any moisture. The relation $\dfrac{CH}{CD}$ is that given by the calorimeter, and is the proportion of dryness at admission.

(2) The drop from C to n indicates a fall of temperature. This is the result of friction and loss of pressure through too small ports or throttling passages.

(3) If it were the case that there were no wire-drawing nor any initial condensation before cut-off, the entropy line

should move to D, and the heat-unit supply should have been greater than it is by the sum of the areas $nicH + HDKLn$. If $niCH$ be charged to wire-drawing, the rest is the result of the condensation before cut-off, or initial condensation. The determination of the point H or n by calorimeter is to this extent unreliable and unsatisfactory, since the initial condensation will mask the other condensation, because greater in amount.

(4) The expansion curve between H and L on the pv diagram becomes the curve nud on the T.E. diagram. If the expansion were truly adiabatic from cut-off, it would descend on the line nL, which is the isentropic line; but since condensation usually continues after cut-off, the curve falls behind the line nL, until abstraction of heat ceases. This occurs where the line nL crosses the expansion line.

(5) Re-evaporation sets in, and a gain in entropy follows from the heating action of the walls. It is here that any action caused by the jackets would also appear.

(6) At d the exhaust opens. If expansion had been complete within the cylinder, d should have been on the temperature level given by the line FG. The expansion at exhaust is into the exhaust-pipe, instead of against the external resistance. The curve of constant volume dc (§§ 201, 202) is drawn, with the attendant loss of area below it.

(7) At c the back-pressure line begins. If c is not on the line FG, it indicates that either the vacuum is not as good as it might be in the condenser, or that the back-pressure in non-condensing engines is unduly high.

(8) At a the exhaust closes, and an area ast of work is done on the cylinder-feed which is a loss.

(9) When the point t is passed, the gain in heat is the result of a warming by the cylinder-walls. The two together may balance each other, although usually the sum is a loss. If the compression of the cushion-steam were just so adjusted that it would fill the clearance volume with cushion-steam at

boiler-pressure, and if the compression were truly adiabatic, the curve *atri* would be the logarithmic curve *OC*. If the compression is insufficient, making *ast* greater than *tri*, the difference measures the net loss.

Hence it will appear that the differences between the area of the actual and the theoretical heat-diagram measure the losses for the whole cycle, and the lost areas under each phase measure the values corresponding to Q_a, Q_b, Q_c, and Q_d in the preceding analytical method.

When the engine is steam-jacketed and an allowance is to be made for the consumption of steam in the jackets, so that their net effect may be observed, the usual plan is to construct a diagram to the right of the ϕ_2 ordinate, having the same T value as the working steam from the boiler, but having a horizontal or entropy length as much less than that of the working steam as the weight of steam per stroke in the jackets is less than the working-steam weight per stroke. This enables the heat-unit area to be compared directly. The jacket-steam parts with its heat energy by contact with the cooler metallic surfaces, as the working steam parts with its heat in non-expansive working when it flows out to the condenser. The lower temperature limit is that of the water condensed in the jacket and removed by traps.

247. Reeves' Entropy-temperature Diagram Chart. —Prof. Reeves arranged in 1897 a most convenient chart for the application of the foregoing principles. Following Boulvin, he divides a large sheet into four quadrants, allotting the lower ordinates below the central horizontal line to pressures, and the ordinates above the central horizontal to entropy. From the central intersection of the horizontal and vertical axes, abscissæ to the right are volumes, and to the left are temperatures. This divides the chart so that, as appears in the diagram (Fig. 108), the four angles give each a diagram in terms of the double unit. In the pressure-temperature segment are drawn a number of different adiabatics for differ-

ent weights of cushion-steam, and in the temperature-entropy segment are the curves of entropy for water and for steam similar to Fig. 106. If then the metamorphosed indicator-diagram be drawn on the $p.v.$ segment with the line AB of Fig. 106 inserted in its proper location, it will be apparent that simple projection of points first upon the proper line in the volume-temperature segment and thence into the T.E. segment will locate the points of the desired experimental T.E.

ENTROPY TEMPERATURE	ENTROPY VOLUME
PRESSURE TEMPERATURE	PRESSURE VOLUME
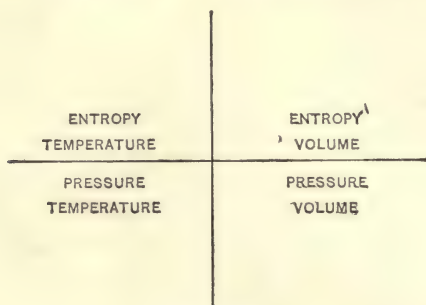
Fig. 108.

diagram. The diagram also presents other data which are intended to make its application more easy.

248. **Conclusion.**—It is to be emphasized anew that these thermal analyses are based on the knowledge as to the quality of the steam with respect to dryness within the cylinder at portions of the stroke, and can be no more accurate than the observations or the assumptions concerning this quality. It is furthermore not always easy to determine with exactness the point of cut-off with single-valve engines, and hence to fix the volume $V_0 + V_1$ for the computations. Hence the conservative attitude towards them is that all tests and experiments should be so directed as to be made available when these disputable facts shall have been settled by an accumulation of knowledge concerning them, rather than that dogmatic assertions can be now made concerning the results of such analysis. These results may be called suggestive rather than conclusive.

CHAPTER XVIII.

COMPRESSED-AIR ENGINES.

250. Introductory.—The foregoing chapters have been mainly concerned with the use of steam as a heat medium, because it is in the first place the most used of all such media, and is the most accessible of the vapor class. Next to steam as a heat medium, the accessibility and harmlessness of air puts it as the most to be preferred of media of the permanent-gas class, treating it as a permanent medium at the usual range of temperatures and pressures.

Before passing to the study of air as a medium to which energy is imparted by the action of heat from combustion, as in the caloric or hot-air engine and in the gas-engine, it will be convenient to refer to a class of air-engines, using air as a motor fluid to which a degree of energy has been imparted by mechanical means so as to raise its capacity for doing work to a level higher than that exhibited by the ordinary atmosphere. The air-compressor (Chapter XIII) is constructed to raise the entropy of air above its atmospheric condition, in isothermal aspiration, and to raise its temperature by adiabatic compression. In so doing a quantity of energy is stored in each pound of air so compressed which is available for work in a piston-motor similar to the steam-engine. The compressed-air engine receives from the reservoir a charge of available energy just as the steam-engine receives its charge from the boiler. A mechanical pressure is exerted against the piston at constant pressure isothermally up to cut-off, with a drop in entropy till cut-off is reached, after which adiabatic

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expansion should drop the temperature in a complete expansion back to where the cycle began. It will be obvious, therefore, that if both compressor and air-motor were perfect and frictionless, the latter could drive the former and together they would make the reversible combination conceived by Carnot. In practice, however, the air-machine is to do some outside work at a point more or less remote from the compressor. The latter has therefore to receive a heat or other energy from some source, transform that energy to the form in which it exists in the compressed air, and permit that stored energy to be transmitted to the remote point, there to be released. It is the convenience and safety of the transmission and storage of energy by compressed air that has made it so important and widespread a feature of modern engineering. Storage of great energy in small bulk and with little weight in strong tanks is the element of strength for compressed air for street-car service or where the motor cannot be conveniently continuously connected to the delivery of the compressor. The convenient return of the exhaust to the atmosphere is in many places an advantage, as underground or in submarine work; and the harmlessness of the air in case of accident, breakage, leakage, and the like, are often valid claims for the use of such air-engines.

Compressed air may be used in air-engines, receiving it from receivers or direct from the compressing cylinders, in three general ways. There can be no condensation with air, so that the lowest pressure level to which it can fall is that of the atmosphere. When it does so, all the mechanical energy is withdrawn from it, and the engine is said to work with complete expansion. The objection to complete expansion is the low terminal effort towards the end of the stroke, which may not be enough to overcome the friction of the motor itself. Hence the second method is that of partial or incomplete expansion where there is a pressure acting at the moment when the exhaust opens. The energy resident in the

air which exhausts is thus lost, but a smaller cylinder will serve to give a given horse-power, because the mean forward pressure is greater, and a lighter fly-wheel will secure a given degree of regularity in speed.

The third method is to deliver air to the cylinder at full pressure, to work without reduction of pressure, and to exhaust a cylinder full of air at the receiver pressure at each stroke. Here the air works without expansion. These three types will be discussed, and the losses in transmitting energy to a distance.

251. Loss of Energy in Transmitting Air through Pipes.—The most considerable loss in well-planned systems will be the loss down the temperature scale by radiation and conduction in the long pipe-line. It does not pay, as a rule, to protect the pipes to mitigate this loss, and furthermore the cooling processes at the compressor are planned to carry away excess of temperature for the sake of lessening the compressor work. As the air cools, however, its volume lessens, or its pressure, or both together, so that the velocity of flow through the long pipe should in theory be increasing slightly from the beginning to the end. Furthermore, to cause a flow of the compressed air in the pipe from the compressor end toward the motor there will require to be a difference in pressure. This may be called $(p_1 - p_2)$ and is a quantity to be assumed by the designer of the pipe transmission. To allow it to be 10 pounds difference is a large value, and would only be justified where the temporary character and small importance of the work made economy of plant-cost of more importance than running or working economy. From 3 to 5 pounds loss in transmissions up to 10,000 feet would not be considered bad practice. It will be apparent, therefore, that length and diameter of pipe will enter the formula, and the density or degree of compression to which the air is brought at the upper or compression end; there must be also an experimental coefficient to embody the actually observed effect

of the condition of the pipe, inside, upon the air flowing through it. The slower the linear flow per unit of time the less effect will the pipe have upon the differences of pressure. Hence it appears that all the factors entering into any formula are likely to be varied by the influence of the others which enter it, making any but an empirical formula a somewhat uncertain dependence.

252. **The D'Arcy Formula for Compressed Air.**—The original formula of D'Arcy for flow of water in hydraulic distributions has been modified to apply to an elastic medium like air, and in its most accepted form appears:

$$D = \frac{c\sqrt{d^5}}{\sqrt{l}} \times \sqrt{\frac{p_1 - p_2}{w_1}}.$$

In this D is the cubic feet of air in cubic feet per minute discharged at the pressure p_2 at the end of the pipe-line when the latter has a length in feet denoted by l and a diameter in inches denoted by d. Therefore the factor $p_1 - p_2$ will be the permitted drop in pressure from the compressor pressure p_1 to secure the demanded final pressure p_2 at the air-engine. The factor w_1 will be the weight in pounds per cubic foot of this compressed air entering the pipe at the compressor or the reciprocal of the volume occupied by one pound at the pressure p_1. Since the weight of a cubic foot of air at 62° F. and atmospheric pressure is .0761 pounds, the weight at any other pressure p_1 will be

$$W_1 = .0761\left(1 + \frac{p_1}{14.7}\right),$$

in which p_1 is in gauge pressure in pounds per square inch. This appears more conveniently

$$W_1 = .0761(1 + 0.068p_1).$$

Values for the factor $c\sqrt{d^5}$ have been worked out as follows:

Diameter of Pipe.	$c\sqrt{d^5}$	Diameter of Pipe.	$c\sqrt{d^5}$
1 inch.	45.3	9 inches.	14872
2 "	297.0	10 "	19480
3 "	876.0	11 "	24800
4 "	1856.0	12 "	30926
5 "	3298.0	13 "	37898
6 "	5273.0	14 "	45690
7 "	7817.0	15 "	54462
8 '	10988.0	16 "	64102

The values for the expression $c\sqrt{d^5}$ in the above table will require to be increased if any unusual conditions as to multiplicity of bends are to be met or the constrictions caused by valves with complicated passages. One bend at right angles has been considered to offer the resistance of one length of pipe of the same calibre. It is here that the judgment of the engineer must come in until exhaustive experiment shall have given reliable figures.

Another experimental formula of considerable acceptance is the result of the practice at the Mont Cenis tunnel excavations. If n denote the velocity in feet per second, d the diameter of the pipe in inches, and l the length in feet,

$$p_1 - p_2 = 0.00936 \frac{n^2 l}{d},$$

when p_1 and p_2 are the pressures in pounds per square inch at the beginning and end of the pipe.

253. Compressed Air-engine with Complete Expansion.—This is the preferred plan, because the air in expanding from the pressure p_3 at which it enters the air-cylinder to the pressure of the atmosphere p_4 at which it goes out, leaves the cylinder without carrying with it any available pressure energy. As in the case of the compressor, the work of the engine will be made up of

$$W_3 = \begin{cases} p_3 v_3 \ldots \ldots \ldots \ldots \text{admission work} \\ + \int_{v_3}^{v_4} p\,dv \ldots \text{expansion work} \\ - p_4 v_4 \ldots \ldots \ldots \text{back-pressure work.} \end{cases}$$

or

$$W_3 = p_3 v_3 + \frac{p_3 v_3}{n-1} \left\{ 1 - \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} \right\} - p_3 v_3 \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}}$$

$$= p_3 v_3 \frac{n}{n-1} \left\{ 1 - \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} \right\},$$

in terms of the initial volumes and pressures for one pound of air received, or if the terminal pressures and volumes be preferred,

$$\bar{W}_3 = p_4 v_4 \frac{n}{n-1} \left\{ \left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}} - 1 \right\},$$

from which the mean effective is derived by dividing through by the final volume v_4, giving

$$\text{M.E.P.} = p_4 \frac{n}{n-1} \left\{ \left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}} - 1 \right\}.$$

In this n is the ratio 1.41 between the specific heats, since $C_p = 1.41 C_v$. The work for M pounds of air would be M times greater. The expansion is considered to be adiabatic, as was the compression.

254. Compressed Air-engine at Full Pressure without Cut-off.—This is a very usual case where the conditions must not permit of the down-drop of temperature in the adiabatic expansion. The air enters full stroke at p_3 and fills a volume V_3, the latter representing M times the volume of one pound at the pressure p_3, when M pounds are expended per unit of time. There is therefore no internal temperature change,

but an isothermal entropy increase during the stroke, and at exhaust the pressure drops to p_4 and the temperature from T_3 to T_4 at the free expansion into the open air, with the loss represented by the product $778(T_3 - T_4)$. In other words, the air is used as an inelastic fluid like water to displace the working piston, and no increase in intrinsic energy resulting from the compression is withdrawn from it in the air-engine. It leaves the latter with just as much as it had when it entered it.

The work of such an air-engine will be the difference between the initial and final pressures into the volume V_3 occupied by the M pounds of air; or

$$W_4 = V_3(p_3 - p_4),$$

but

$$p_3 v_3 = MRT_3;$$

hence

$$W_4 = MRT_3\left(1 - \frac{p_4}{p_3}\right)$$

$$= 778M(C_p - C_v)T_3\left(1 - \frac{p_4}{p_3}\right),$$

since by definition $C_p - C_v = \dfrac{R}{778}$ (§§ 116 and 183).

To compare this with the work of complete expansion, the expression for the work of one pound is transformed by the relations

$$C_v(n - 1) = \frac{R}{778}.$$

Hence, since $p_3 V_3 = RT_3$, and

$$\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}} = \frac{T_4}{T_3},$$

the expression

$$W_s = p_s V_s \frac{n}{n-1}\left[1 - \left(\frac{p}{p_s}\right)^{\frac{n-1}{n}} \right]$$

can be written

$$W_s = 778 C_p T_s\left(1 - \frac{T_4}{T_s}\right)$$

for one pound of air; or for M pounds

$$W_s = 778 M C_p (T_s - T_4).$$

This expression also gives the temperature range for any assigned work given in pressure relations, with complete expansion. The work at full pressure may therefore be conveniently equated to an expression of the above form in which an unknown temperature T_x shall replace the final temperature T_4 in this last equation. That is,

$$W_4 = 778 M(C_p . C_v)(T_s - T_4) = 778 M C_p(T_s - T_x).$$

which will express that final temperature giving the same work in a complete expansion as was given by the full-pressure condition. Solving for T_x it becomes

$$T_x = \left[\frac{1}{n} + \frac{n-1}{n} \cdot \frac{p_4}{p_s} \right] T_s ,$$

whence the ratio between T_s and T_x becomes when the numerical value for n is inserted:

$$\frac{T_x}{T_s} = 0.71 + 0.29\frac{p_4}{p_s}.$$

T_x is a temperature having no experimental or actual value; but from it the relations of the work done by complete expansion and full pressure for any values of the ratio $\frac{p_4}{p_s}$ can be worked out.

255. Compressed-air Engine with Incomplete Expansion.—The third case is perhaps the most usual, where air is received at a pressure p_3, volume V_3, and temperature T_3 up to cut-off; is expanded adiabatically down to conditions p_4, V_4', and T_4' above the exhaust conditions represented by p_4, V_4, and T_4. The air expands freely from p_4' down to p_4, doing no external work, but there has been some lowering of the initial pressure in doing work during the partial expansion. As before,

$$W_5 = \begin{cases} p_3V_3 \dots\dots\dots\dots\dots\dots & \text{admission work} \\ + 778MC_v(T_3 - T_4') \dots & \text{expansion work} \\ - p_4V_4' \dots\dots\dots\dots\dots & \text{back-pressure resistance.} \end{cases}$$

Following the same plan as in the preceding case, and substituting an ideal temperature T_x in an equation of the form given with complete expansion which shall give the same work at full expansion as is given in the actual case of incomplete expansion, the above expression for W_5 will be placed equal to the ideal expression, and solved for T_x. That is,

$$W_5 = p_3V_3 + 778MC_v(T_3 - T_4') - p_4V_4' = 778MC(T_3 - T_x).$$

But since

$$V_3 = 778M(C_p - C_v)\frac{T_3}{p_3},$$

and

$$V_4' = 778M(C_p - C_v)\frac{T_4'}{p_4'},$$

this becomes when divided through by $778M$

$$(C_p - C_v)T_3 + C_v(T_3 - T_4') - (C_p - C_v)T_4'\frac{p_4}{p_4'} = C_p(T_3 - T_x).$$

whence by performing operations,

$$C_v T_4' + (C_p - C_v) T_4' \frac{p_4}{p_4'} = C_p T_x,$$

which gives

$$\frac{T_x}{T_4'} = \frac{Cv}{C_p} + \frac{Cp - Cv}{Cp} \frac{p_4}{p_4'} = \frac{1}{n} + \frac{n-1}{n} \frac{p_4}{p_4'} = 0.71 + 0.29 \frac{p_4}{p_4'},$$

which is the same ratio as found for the non-expansive working. Hence for both cases

$$W_4 = 778 M C_p (T_3 - T_x) = 778 M C_p T_3 \left(1 - \frac{T_x}{T_3}\right),$$

and

$$W_5 = 778 M C_p T_3 \left(1 - \frac{T_x}{T_4'} \frac{T_4'}{T_3}\right).$$

The values of T_x can be found for any relation of p_4 and p_4' from the foregoing identical formulæ.

256. Compressed-air Engine with Isothermal Expansion.—This is a very unusual case, because it means heating the working air so as to have the same terminal temperature as at the entry, by some hot jacket or similar device. It is conceivable, however, if the high temperature of exhaust were to be thought desirable.

The work per stroke will be:

$$W_7 = \begin{cases} p_3 v_3 \ldots \ldots \ldots \text{admission work,} \\ \displaystyle\int_{v_3}^{v_4} p\, dv \ldots \text{expansion work,} \\ -p_4 v_4' \ldots \ldots \text{back-pressure work,} \end{cases}$$

when expansion is incomplete, and

$$W_8 = \begin{cases} p_3 v_3 \ldots \ldots \ldots \text{admission work,} \\ \displaystyle\int_{v_4}^{v_3} p\, dv \ldots \text{expansion work,} \\ -p_4 v_4' \ldots \ldots \text{back-pressure work,} \end{cases}$$

when expansion is complete.   Hence

$$W_1 = p_3 v_3 \left[1 + \text{hyp. log } \frac{v_3}{v_4{}'} - \frac{p_4}{p_4{}'} \right],$$

and

$$W_3 = p_3 v_3 \text{ hyp. log } \frac{v_4}{v_3} = p_3 v_3 \text{ hyp. log } r,$$

when r is the ratio of expansion, or the ratio between the volumes at cut-off and at release.   With complete expansion the terminal pressure $p_4{}'$ at end of expansion equals the back-pressure p_4, or $\frac{p_4}{p_4{}'} = 1$.

257. Volume of the Cylinder of a Compressed Air-engine.—The design of an air-engine cylinder usually presents itself with the air-pressures given, the ratio of pressures at beginning and end, and the foot-pounds or horse-power of work to be done.   Hence the formula of §§ 168 and 181 is directly available.   If the known horse-power reduced to foot-pounds is divided by $2n$, when n is the number of revolutions per minute, the quotient will be the work to be done in one stroke.   Substituting this for W_3 and solving for v_4, the necessary final volume of cylinder is found, neglecting clearance loss.   The final volume will be the product of the two factors area $\times$ stroke, which must be proportioned to each other according to any determining conditions as to either.

Or, the mean pressure value (§§ 184, 253) can be substituted in the equation

$$\text{H.P.} = \frac{PLAN}{33000}$$

for the factor P, and the equation solved for LA.   Clearance will increase the cylinder volume according to the data of §§ 233–236.

258. Compound Compressed-air Engine.—The use of a non-condensing type of engine is forced upon the designer of

an air-engine. Hence when large powers are to be stored in small bulk the use of high pressures is made necessary, and, with a fixed lower pressure limit, a large ratio between $\dfrac{v_4}{v_3}$. To secure this high degree of expansion or early cut-off in a single cylinder forces the use of disadvantageous crank-angles for the admission pressures, making, therefore, unsatisfactory working of the engine from the great range of pressures over the area of the piston. This evil is much mitigated by the use of the multiple-expansion system, using cylinders of progressive areas or volumes as the pressures fall during expansion, and enabling each cylinder to have a longer and more advantageous admission (§§ 172–174). The difficulties from cylinder condensation, which are of such moment in the steam-engine, are of less moment in the air-engine and may be disregarded, although there is an interchange of heat with the metal walls and the working fluid. Hence the work of the entire expansion from p_3 to p_4 is divided equally between the two, three, or four cylinders of the series by laying down a diagram having an area equal to the whole work under the assumed degree of expansion supposed to take place in one cylinder, and then dividing this area of work into halves, thirds, or quarters, and giving to each cylinder in the series a volume proportioned to the pressures within which it works.

The compound or multiple system permits a reheating between the cylinders if desired, whereby unpleasantly low terminal temperatures may be mitigated and the expansion be brought nearer to the greater work of the isothermal curve of pressures and volumes. That is, if there be two cylinders, and the terminal pressure and temperature in the first cylinder be p_4' and T_4'. Hence

$$W_2 = 778 C p (T_3 - T_4') = 778 C p T_3 \left[1 - \left(\frac{p_4'}{p_3} \right)^{\frac{n-1}{n}} \right].$$

If in an intermediate receiver, by any source of heat, the expanded air is raised again to T_3, the terminal temperature will become T_4'' and the work in the second cylinder will be

$$W_{10} = 778 C p (T_3 - T_4'') = 778 C p T_3 \left[1 - \left(\frac{p_4}{p_4'} \right)^{\frac{n-1}{n}} \right],$$

so that the sum of the two effects will be

$$W_9 + W_{10} = 778 C p T_3 \left[2 - \left(\frac{p_4'}{p} \right)^{\frac{n-1}{n}} - \left(\frac{p_4}{p_4'} \right)^{\frac{n-1}{n}} \right].$$

The work will be a maximum when the last two terms are a minimum. This occurs when

$$p_4' = \sqrt{p_3 p_4},$$

as was the case with the compound compressor (§ 189). The question, however, is not yet answered by experiment, whether the loss in clearances and free expansion drop be-
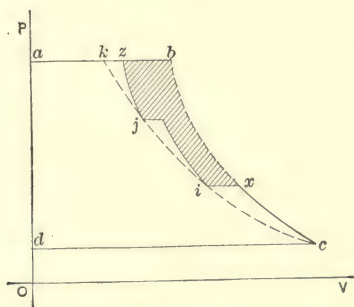
Fig. 109.

tween the cylinders, when added to the interest cost of the smaller cylinders, offsets the mechanical gain from the use of the additional cylinders. In street-car motors, where two cylinders are wanted in any event to prevent stalling on

centres, the compound method offers great advantages. It will be interesting to compare the diagram of the advantage from intercooling in compression of air (Fig. 109) with the increase of volume resulting from interheating between cylinders which causes an adiabatic expansion in the air-engine to approximate more closely to the isothermal condition.

259. Combined Efficiency of Compressor and Air-engine.—Since the compressed-air engine can only be operated in connection with a compressor which has antecedently raised the pressure and temperature level of the air, it becomes of moment to compare the work of the two machines in the light of the foregoing discussion, so as to reveal the directions for effort to make the air-engine return in mechanical energy all the energy put into the compressor by the prime source of power. Since the temperatures are the significant factors, the work of the compressor will be transformed from the expression in § 169 by the method followed in § 254, whence

$$W_1 = 778 C_p (T_2 - T_1)$$

The work of the air-engine will be

$$W_3 = 778 C_p (T_3 - T_4).$$

The efficiency, being the quotient of the delivered work by the applied work, becomes

$$E = \frac{W_3}{W_1} = \frac{778 C_p (T_3 - T_4)}{778 C_p (T_1 - T_2)} = \frac{T_3 \left(1 - \dfrac{T_4}{T_3}\right)}{T_2 \left(1 - \dfrac{T_1}{T_2}\right)},$$

which can be transformed into pressure relations by writing

$$E = \frac{T_3 \left(1 - \left(\dfrac{p_4}{p_3}\right)^{\frac{n-1}{n}}\right)}{T_2 \left(1 - \left(\dfrac{p_1}{p_2}\right)^{\frac{n-1}{n}}\right)}.$$

If the compressor and air-engine operate with about the same ratio or range of pressures, the ratio of the bracketed factors is not far from unity. This makes it appear that the nearer the temperature of inlet into the air-engine approaches the temperature of the delivery from the compressor, the nearer will the efficiency become unity. Hence the wisdom of preheating the air for the air-engine, if the latter is at any distance from the motor so as to have lost any of its higher temperature T_2.

It must not be forgotten, however, that it is not mechanically possible to reach an efficiency of unity, even with preheating at the air-engine, if the work of the engine part of the compressing plant be taken as the starting-point. If the compressor return to the air 80 per cent of the work put into the compressor, and the air-engine deliver 80 per cent of the work which it received, the double transmission and transformation returns at the air-engine as its net work only 80 per cent of 80 per cent of the steam-cylinder work, or only 64 per cent, even with complete expansion, unless the efficiency of the air-engine can be made greater than unity by adding extra heat energy at the point where the engine works.

If T_3 is the temperature of the atmosphere at the air-engine, it would be an advantage to lower T_3 by cooling or otherwise. This confirms the advantage from isothermal compression or two-stage compression when the loss of energy in the cooling water is of less moment than the other compensating gains.

260. Heat Range in the Air-engine Cylinder.—The expression for the work of the air-engine,

$$W_3 = 778 C_p (T_3 - T_4) = 778 C_p T_3 \left[1 - \left(\frac{p_4}{p_3} \right)^{\frac{n-1}{n}} \right],$$

leads at once to the conclusion that if the air enter the air-engine at atmospheric temperatures—between 60° and 68° F.

—it will leave it at very low temperatures if there is any considerable range of pressures. The following table gives values for T_4 absolute and Fahrenheit calculated from the relation

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{n-1}{n}},$$

assuming

$$T_3 = 459.4 + 68°.$$

$\frac{p_3}{p_4}$.	Final Absolute Temperature t_4.	Final Fahrenheit t_4.	$\frac{p_3}{p_4}$.	Final Absolute Temperature t_4.	Final Fahrenheit t_4.
2	431.4	− 28	9	278.9	− 180.5
3	383.5	− 75.9	10	270.5	− 188.9
4	352.8	− 106.6	11	263.1	− 196.3
5	330.7	− 128.7	12	256.6	− 202.9
6	313.7	− 145.7	13	250.7	− 208.7
7	299.98	− 159.4	14	245.3	− 214.1
8	288.5	− 170.9	15	240.5	− 218.9

These values for the temperature of the exhausted air lead to the consideration and feasibility of mechanical refrigeration, which will be considered in a subsequent chapter, and also to the signal advantage of preheating the air before it enters the air-engine. It further makes clear the objection to the presence of moisture in the air used, because such moisture becomes snow or ice in the exhaust-passages and clogs them.

261. Preheating the Air for the Air-engine.—Large-scale experiments with compressed-air plants without preheating of the air at the motor show an efficiency ranging from less than 30 per cent up to about 40 per cent. Preheating may reduce the losses to something over 20 per cent only, realizing an efficiency between 70 and 80 per cent. That is, to heat the air to a temperature of 480° F. at the motor will result in an increase of efficiency of 30 per cent.

Methods of heating the air involve either some form of stove, or the use of hot water under pressure. The stove

plan in its simplest form is to force the compressed air through an air-tight vessel in which anthracite coal or charcoal is kept incandescent by the union of the carbon and the oxygen of the compressed air. The trouble with this arises from the ash and dust of the fuel going forward through the engine. For the small compressed-air motors of Paris, a stove of cast iron lined with fire-clay is heated by a gas-jet or a small coke fire. A coil of pipes in a fire forms another type. The fuel consumption is so small as to be scarcely noticeable, or about 0.2 of a pound of fuel per horse-power per hour. Seventy per cent of the available heat in the fuel went into the air, raising its temperature from 170° to 300° F. above the temperature in the conducting pipe.

A form of preheater which has been used for street-car service causes the compressed air to pass through a pressure-tank filled with superheated water at 330° F. This avoids carrying live fire on the car. The water carries more heat per unit of weight than any other body, and the air, taking up some water mechanically, causes it to become vapor in the cylinder, adding to the propelling effect. This is the feature of the Mekarsky system. Other plans inject hot water in jets into

FIG. 110.

the air storage reservoir. It will be apparent that preheating raises the value of T_3 to that of T_2 in the formula of § 259 and tends towards an efficiency of unity. Fig. 110 shows a cut of such preheater using oil.
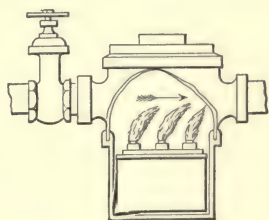
262. Temperature-entropy Diagram for Compressed-air Engine.—While the air-engine acts like the non-condensing steam-engine in one sense, yet it is only possible to treat it satisfactorily in connection with the compressor, of which it is the complement (§§ 263 and 264), since the cycle is not closed otherwise. If, for example, the perfect-expansion *pv*

diagram be chosen (Fig. 115), the admission of air at constant
pressure from the receiver ab is not an isothermal at the
temperature T_s because the pressure is not maintained by
adding heat to the reservoir; but if the latter is of finite
volume, the pressure is maintained by the inlet-supply from
the compressor. Hence the line on the temperature-entropy
diagram (Fig. 116) will be a line of constant pressure
descending by a logarithmic curve from the point b to a for
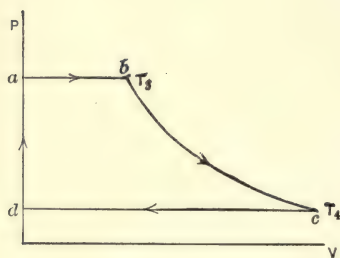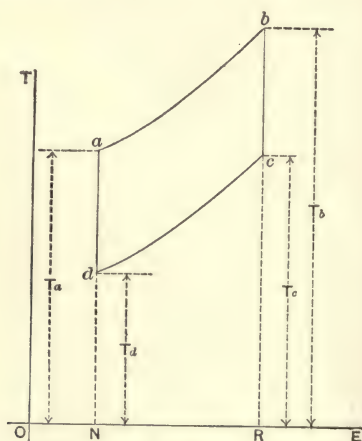
FIG. 115.   FIG. 116

which the upper limit (T_b or T_s) is the temperature of the air
leaving the compressor, and the lower limit is the tempera-
ture (T_a or T_s) at which after cooling by radiation the air
enters into the cylinder of the air-engine. Hence the rela-
tion of the co-ordinates will be

$$\phi = C_p \text{ hyp. log } \frac{T_b}{T_a}.$$

At the point a the air begins to expand in the air-engine
proper (it will be observed that the line ab really belongs in
the compressor process), and by hypothesis such expansion

is adiabatic, and is accompanied by a drop in temperature without entropy change, giving the line *ad*, corresponding to the expansion-line *bc* on the *pv* diagram. In the absence of preheating (§ 261) T_d is much lower than the atmospheric temperature with usual points of cut-off in the air-engine, so that T_d gives a point *d* (§ 260) lower than the air into which the exhaust escapes. Hence a constant pressure-curve *dc* is required to return the exhausted cold air to the normal atmospheric condition, and its equation will be

$$\phi = C_p \text{ hyp. log } \frac{T_c}{T_d}.$$

It will be apparent that the adiabatic compression of the air-compressor will be required to close such a diagram by a compression from T_c up to T_b from which the process began. If, however, the process be an incomplete expansion, as in the dotted *pv* diagram of Fig. 2, or without any expansion whatever, as in Fig. 1, the temperature-entropy diagram will appear like the left-hand part of Fig. 117 for the first case, and the right-hand part for the second.
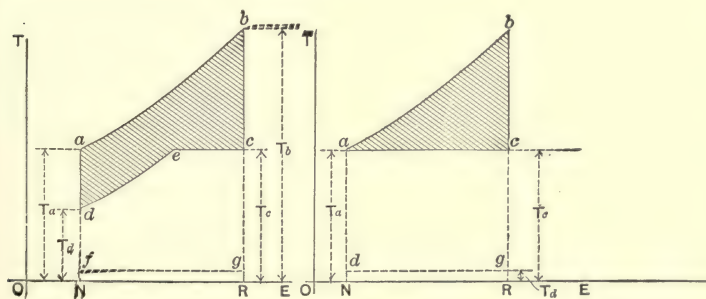
FIG. 117

The greater relative area of rejection of heat in these latter cases is the measure of their less economy as com-

pared with the full-expansion class, provided that the greater capacity of the working cylinder for a given power by reason of the lower mean pressure is not an offset in part to the greater heat-rejection per stroke.

263. Temperature-entropy Diagram for the Air-compressor.—The compressing cylinder *cd* being full of air at atmospheric pressure T_1 (Fig. 118), the return of the piston
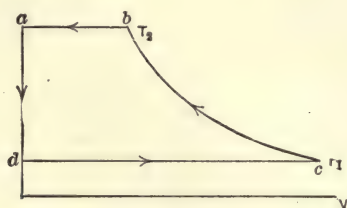
Fig. 118.

compresses air adiabatically up temperature along the vertical ordinate *cb* (Fig. 116), without change of entropy to the pressure and temperature belonging to T_2. Then the valves of the receiver open in the actual case and the air passes at constant pressure by slow discharge of its heat energy down the curve of constant pressure to the condition as to temperature which belongs to surrounding objects. This is the line *ba* of Fig. 118, and may logically attach itself to either the compressor or the air-engine. Strictly, of course, it must be supposed to occur in the compressor-cylinder, after the adiabatic compression is complete to the pressure p_2. Then, in the absence of any adiabatic expansion and drop down temperature, the diagram should close by an entropy change at constant temperature $T_a = T_c$ so as to be capable of compression again to the point *b*.

If the cooling is not permitted at the constant pressure p_2 of the receiver, then the increase of entropy value does not occur, and the diagram becomes the straight isometric *cb*, first up, and then down. This represents a cycle of pure

adiabatic type, with full storage and restoration of the heat energy of the compression, so that the air finishes with the same energy that it began with, and acts like a spring. No effective outside work has been done, however, because no energy has been expended. $Td\phi$ has a zero value.

If, on the other hand, the air reached the air-engine at T_b by an isothermal heating process, the point a would lie on a horizontal through b, and the adiabatic expansion through the temperature-range represented by $ad = cb$ would bring the final temperature to that of the intaken air, and the exhaust in escaping would retain sufficient energy to return by a constant pressure-curve up again to b, if it were not cooled by outside means. In practice, of course, the exhausting air is cooled by contact with the atmospheric air, and this potential energy is lost.

264. **Temperature-entropy Diagram for the Combined Air-compressor and Air-engine.**—A much more valuable conception and application of the heat diagram results, therefore, from the consideration that the compressor and the engine which it drives are one apparatus, the exhaust from the air-engine forming the aspiration volume of the compressor, and the discharge from the latter forming the admission volume for the stroke of the air-engine. The compressor always works with complete expansion; the air-engine must be of the same capacity and work with complete expansion down to the intake-pressure. The object is to have no loss in transmission, and the external work done by the air-engine should be the same as that done by the steam-engine cylinder of the compressor. Then the constant pressure-line and the adiabatic at the higher entropy belong to the air-engine, and the lower constant pressure-line and the adiabatic at the lower entropy are the complementary parts of the compressor cycle. The atmospheric ocean is the receiver for the air exhausted from the air-engine, and delivers it without change of state to the compressor-inlet.

The extent to which the actual combination departs from this ideal complete cycle made up of their partial diagrams measures the heat losses of efficiency, by reason of the failure of the entropies to equalize, or because a temperature loss or
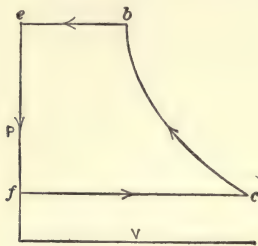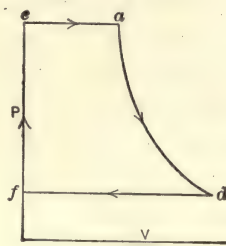
FIG. 119.　　　　　　　　FIG. 120.

a pressure loss by cooling and radiation has compelled an expenditure of energy at the compressor greater than that exhibited by the air-engine.
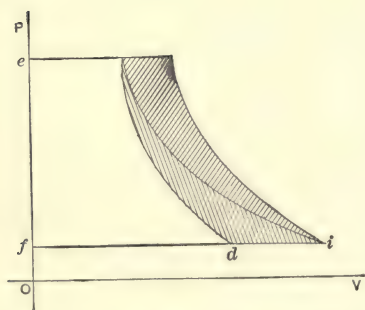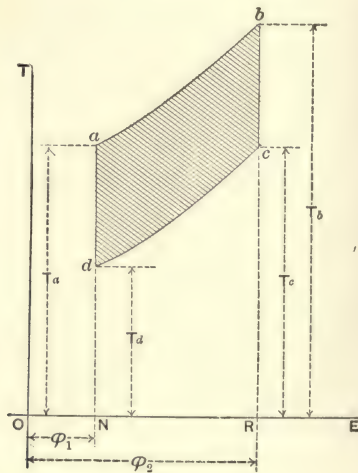
FIG. 121.　　　　　　　　FIG. 122

Let it be supposed, for example, that the *pv* diagram (Fig. 119) is the card from the compressor, and Fig. 120 is

the adiabatic card from the complementary air-engine. The compression of the air is adiabatic, and after passing through pipes and storage its volume at the air-engine is reduced, as is made clear by the superposition of the two cards (Fig. 121). The shaded area lying between the two adiabatics is the loss of work as revealed by the pv representation of it. The temperature-entropy diagram in Fig. 122 deduced from the two foregoing paragraphs and resulting from their superposition shows that the heat energy furnished by the compressor is the area $baNR$, and the energy rejected by the air-engine is represented by the unshaded area $cdNR$. Hence the available or utilizable energy should be their difference, or the shaded area $badc$.

The effect of preheating if carried so far as to raise the point a to the same temperature-level as b is made evident by an increase in the area denoting energy supplied, enabling a greater degree of adiabatic expansion if the same temperature-level d is fixed, or a higher value for d can be permitted, provided the designer is willing to increase the area of rejected energy for the same work in the air-engine cylinder. If now, on the other hand, the compression were isothermal, and the air by the abundant use of cooling water were not allowed to rise in temperature, the temperature entropy diagram will be a line through c parallel to the entropy axis, since $T_2 = T_1$, and the receiver is supposed to be large enough not to have its temperature raised by the displacement of the compressed air into it. This means, then, that the cooling water has carried away with it (perhaps to waste) a quantity of heat energy equal to the work of the compression, and the compressed-air receiver has taken up the work of displacement by molecular movement and eddies in the air itself. When, then, the air-engine receives its air at atmospheric temperature from such an isothermal compression, and uses it in complete adiabatic expansion, the air

must drop down temperature on expansion by an amount which would cool that weight of cooling water back to its original temperature if it could be kept for this purpose. If the water has been wasted, then the universe at large has received the heat from the cooling water, and has to supply that which warms the exhaust-air up to T_1 again. The heat of the fuel burned to compress the air in the first place is wastefully used in this process, when referred to the power developed at the air-engine.

Or, again, if the expansion at the air-engine were made isothermal by heating the air as it was working as well as before admission, making $T_3 = T_4$, the temperature-entropy diagram becomes a line also, and the heat to be supplied will be that which will be the area of the finite diagram which would have been generated with true adiabatic expansion.

These observations lead to the generalization which might have been foreseen in advance, that compressed air or other elastic media are only effective storage media for mechanical energy, and act like a spring, provided that means are taken on the expansion to replace the heat conditions in inverse order which attended the compression. Or in other words, air acts as water or other incompressible fluid transmission would to transmit the energy of the steam or other motor, if pains are taken to prevent either uncompensated rise or fall of temperature while in use as a medium : and in this case friction in pipes, valves, and bends would be the only source of loss.

Finally, if heat is supplied to the air-engine by preheaters, such heat, adding to that received from the compressor, adds an area to the temperature-entropy diagram which helps to compensate for the losses of energy which will appear in actual conditions.

265. Concluding Summary.—It must not be overlooked that the air-engine and the air-compressor are not heat-engines in the sense in which this treatise uses this term. They do

not create a mechanical energy or liberate it from a reservoir of stored heat energy. The mechanical energy being created or liberated so as to be available, these appliances store and restore that mechanical energy. Their claim for consideration is based upon the fact that the relations of pressure volume and temperature are so interrelated by natural laws that the principles underlying the compressor and compressed-air engine must conform to those broader and more fundamental ones which the science of thermodynamics must consider as its own. The special field of compressed-air engineering bears, however, to the general subject of motive-power engineering the same relation which is borne by the electric dynamo and motor in its parallel department : the energy having been liberated or made available, each is a convenient and satisfactory method of transmitting that energy to desired points and utilizing it there. The economic advantages of large-scale installations for the generation of mechanical energy are reaped by either system, and the choice of that which is to be preferred must often be guided by considerations outside of the purview of pure theory.

CHAPTER XIX.

HOT-AIR ENGINES.

266. Introductory.—In the preceding chapter a class of engines using cool or cold air has been discussed, in which the propelling energy to drive the piston was given by raising the pressure of air as a medium by means of mechanical energy previously available. This chapter is to discuss air-engines in which the pressure to drive the piston is given to the air as a heat medium, by increasing its temperature and entropy directly by the application of heat to it. They may or may not include also the conception of a compression of that air, but the rise of its temperature is the primary feature.

The relatively low specific heat of air:

at constant volume............0.16847, (C_v)
and at constant pressure........0.23751, (C_p)

together with the reluctance with which heat is transferred to it, except by contact in thin films or small subdivided masses, not only limit the size and weight of these engines to relatively low powers, but, by making the storage process inconvenient between the source of heat and the engine, the boiler and its appendages, have been thrown out which form so considerable a feature of the steam-engine and constitute one of its dangers. The usual underlying principle is the heating and cooling of

the air for each stroke between the limits of p_2 and p_1 in the cylinder itself, or in small chambers without it, or both. Carnot's principle applies most closely to such engines, because the same weight of air is often used over and over again in a strictly closed cycle. The term "caloric" engine is a sort of trade name applied by John Ericsson in 1833 to his engines of this sort, and is properly going out of use.

267. Types of Hot-Air Engine.—The fact that air as a heat medium is also a supporter of combustion, and the high temperature directly realizable from such combustion, has attracted designers to the use of engines in which the heat energy should be directly liberated in the working cylinder. Such engines will be called *internal-combustion* engines, the furnace being practically within the cylinder or in communication with it. The engine works rather with the products of combustion than with air as a medium, and for this reason such engines are sometimes called "products of combustion" engines. The fuel may be solid, liquid, or gaseous (Chapter V). The importance of the gas-engine will command a subsequent chapter for itself; the oil-engine makes the fuel gaseous before it is consumed, and is in the gas-engine class.

The alternate plan is to have the furnace exterior to the working cylinder, heating the working medium by transfer through a metal wall. This is the more usual type where the fuel is solid, and these form the hot-air engine properly so called.

A second classification of type must be made which shall include in one class those engines which operate their cycle upon the same mass or weight of air continuously, only taking in a fresh charge to replace leakage losses or to increase the mass in use. This type might be called the closed-cycle type. The other class in this division would be the open-cycle type, where at each stroke a new charge is drawn in from the atmosphere

and, after being heated and expanded, is exhausted again into
the atmosphere, as occurs in the non-condensing steam-engine.
The first class can evidently be operated independent of atmos-
pheric pressure, or with an initial tension selected at pleasure.

Air-engines will again differ according as they use or do
not use the principle of the "regenerator" to absorb heat on
the outflow of the air, and to restore such entrapped heat to
the incoming cooler air.

Finally, engines of the closed-cycle type may differ by hav-
ing the temperature change in the air take place at constant
pressure or at constant volume. Each type is identified with
the name of some designer or engineer. It will be apparent
that the closed-cycle hot-air engines operate on the lines of the
Carnot cycle, as the steam-engine can only be made to do by
making the feed-pump or air-pump an integral part of the
series of organs.

268. Regenerator for Hot-air Engine.—The desirability
of an appliance within a heat-engine itself which could absorb
heat on its way to rejection at T_1 and give it up to the air on
its way to become heated up to T_2 was early realized by stu-
dents of the Carnot theorem. If it could be made to work,
such an appliance could replace some of the necessity for
adiabatic expansion and compression (particularly the unde-
sirable latter). The first to apply the idea was Robert
Stirling (1816 and 1827), and Ericsson used it in 1833 and
thereafter. The ideal plan would be to have a chamber of
such thermal capacity, and such absence of self-conductivity,
that the air entering it at T_2 at its cool end should be heated
to T_1 by the time it had reached the other or hot end ; and
such heat as was thus absorbed should have been imparted to
the chamber by the flow of the hot air entering in the reverse
direction at a temperature T_1 so as to leave the chamber at the
cooler end at T_2. This hypothesis is of a distinctly reversible

process, but fails of course from the actual conduction and radiation, and from the necessarily limited capacity for heat of a regenerator of practicable size and weight.

These regenerators as applied to hot-air engines are either wire-gauze nettings or thin brass plates on edge for small engines, or are coils or grids of copper wire for larger ones. The Ericsson hot-air ship of 1853 had a wire regenerator whose aggregate length of wire exceeded fifty miles. The weight of material used in the regenerator in British practice seems to have been about forty times the weight of air used per stroke. The waste from practical causes seems to have ranged from one tenth to one twentieth of the heat alternately withdrawn and restored per stroke. The closed-coil feed-water heater using exhaust-steam embodies this regenerator idea.

It must not be overlooked that the action of the ideal regenerator has nothing to do with the taking in and rejection of heat by the engine for the purpose of doing work with that heat. The regenerator becomes thus an integral part of the engine, and is to be so considered.

269. Hot-air Engine with Temperature Changes at Constant Volume. Stirling's Engine.—The engine designed by Robert Stirling (1816) and improved by James Stirling (1827) is one of the simplest of the closed-cycle external-furnace type, and embodies regenerator and Carnot cycle. It is shown in ideal section in Fig. 125.

The working cylinder is *B*, with its piston connected to a beam and so to the rotative mechanism. A passage or pipe connects the working cylinder with the larger chamber in which the heat changes occur. *D* is a displacing piston or plunger made of a sheet-metal casing filled with a non-conducting material like plaster or brick-dust. The furnace-fire surrounds the hemispherical cast-iron vessel at the bottom,

but is prevented from reaching above the diameter. In the
upper part at *C* are coils of small copper pipe in which circu-
lates cooling water forming the refrigerator. Between the
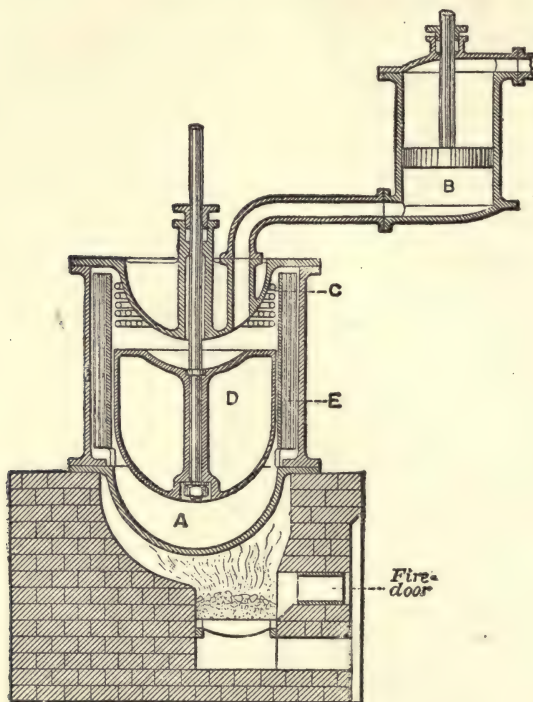
Fig. 125.

points *A* and *C* is the regenerator *E* of thin plates. The
displacing plunger *D* does not fit the casing, but an inner
lining of it, which lining is perforated at the bottom. The
working air is thus forced to pass through the regenerator on
its way to and from the working cylinder. The displacing
plunger *D* is so adjusted as to its phase of movement that the
up-stroke shall occur when the working piston in *B* is down
or nearly so, and the descent of *D* shall take place when *B* is

near the beginning of its idle or descending stroke. The
engine is single-acting unless made
with double cylinders. The cycle
presented on the pv diagram is there-
fore shown by Fig. 126.

(1) The air at T_1 from the action
of the regenerator receives heat at
that temperature from the furnace. It
expands isothermally along ab, doing
work through the piston B. The
heat taken in per pound of air will be
$C_p T_1$ hyp. log r, in which r is the

FIG. 126.

ratio between the initial and final volumes of the air. The
increase in entropy during that isothermal expansion will be

$$\phi_2 - \phi_1 = R \text{ hyp. log } r.$$

(2) The air returns without change of its volume through
the refrigerator from the T_1 end to the T_2 end. The pressure
fails along bc proportionally to the fall in temperature. The
heat stored in the regenerator plates is $C_v (T_1 - T_2)$.

(3) The ascent of the displacing plunger compresses the
air along cd, reducing its volume, but withdrawing any heat
by the passage over the coils of the refrigerating surface in
DC. Heat is rejected to the cooling water to the amount of
$C_p T_2$ hyp. log r. The entropy change should be the same as
in the expansion, but in reverse direction.

(4) The air, passing again through the regenerator from the
T_2 end to the hot end, is heated to T_1 isothermally at constant
volume along da, and the heat taken up by the air should be
that furnished to the regenerator on the other transit of the
air, or $C_v (T_1 - T_2)$.

**270. Temperature - entropy Diagram for a Stirling
Hot - air Engine.**—If the regenerator were ideally perfect,
and replaced the adiabatic expansions and compressions per-

fectly, the temperature-entropy diagram would be the rectangle whose width was (Fig. 127)
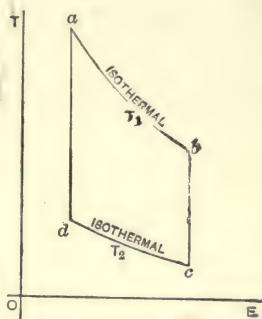
$$\phi_2 - \phi_1 = R \text{ hyp. log } r$$

and whose length was

$$T_1 - T_2,$$

and whose area

$$(\phi_2 - \phi_1)(T_1 - T_2)$$

would give the mechanical energy per pound of air per stroke. The relation of this quantity to the applied heat,

$$(\phi_2 - \phi_1)T_1,$$

would be the efficiency.   The rejected heat would be

$$(\phi_2 - \phi_1)T_2.$$

But while the regenerator gives when perfect in action a temperature-entropy area equal to that of the Carnot cycle since,

$$\frac{CT_1 \text{ hyp. log } r - CT_2 \text{ hyp. log } r}{CT_1 \text{ hyp. log } r} = \frac{T_1 - T_2}{T_1},$$

the diagram is not of precisely the same form, because there will be a gradual decrease in entropy during the cooling, and a
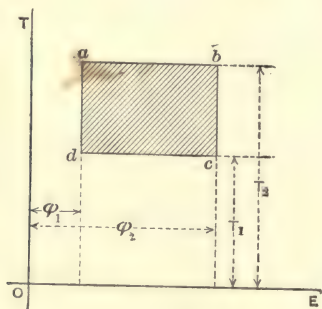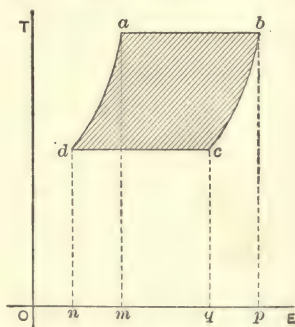
FIG. 127.   FIG. 128.

gradual increase during the heating process.   Hence the diagram will take the form of Fig. 128.   The curve *bc* replaces

the isentropic drop in temperature, and the curve da represents the corresponding rise. If the regenerator is perfect, these two curves will be similar, because each transfer in one direction is the same as the transfer between the same temperature limits in the other. The value for the abscissa at any temperature T will be

$$\phi = C_v \text{ hyp. log } T$$

when the transfer is at constant volume, as in the case under consideration. For the other class the specific heat will be that at constant pressure.

The area $pbcq$ shows the heat taken in by the regenerator, and the area $madn$ shows the heat given out to the air.

Taking the actual experimental data for a Stirling air-engine used many years ago in a foundry at Dundee, and quoted from the Institute of Civil Engineers of Great Britain, some interesting conclusions may be drawn. The observed quantities were

$$T_1 = 1111 \qquad p_1 = 240 \qquad d = 16 \text{ inches}$$
$$T_2 = 611 \qquad r = 1.25 \qquad l = 4 \text{ feet}$$
$$Rpm = 28$$

Volume of cylinder
   at admission and end of exhaust $\quad = 1.709$
   at end of expansion and beginning of exhaust $= 2.119$

Expenditure of heat in heating the air, or latent heat of expansion,

$$H_1 = P_1 \overset{T_1}{\phi} = R \text{ hyp. log } r \times \overset{T_1}{P_1} = 11.647 \times 1111 = 12942$$

Waste heat in regenerator,

$$mK_v(T_1 - T_2) = 13 \times 500 = \quad 6500$$
$$(m \text{ is called from } \tfrac{1}{10} \text{ to } \tfrac{1}{20})$$

Total heat expended per pound per stroke $\qquad\qquad \overline{19442}$

Rejected heat, $H_2 = T_2\phi = 11.647 \times 611 = 7119$

Mechanical energy per stroke per pound,

$$\phi(T_1 - T_2) = 11.647 \times 500 = 5823$$

Efficiency of fluid,

$$\frac{\text{Mechanical energy}}{\text{Heat applied}} = \frac{5823}{19443} = 0.3$$

Volume of piston displacement per stroke,

$$\frac{2.119}{2} = 1.06 \text{ cubic feet.}$$

Mean effective pressure,

$$\frac{\text{Mechanical energy}}{\text{Volume of displacement}} = \frac{5823}{1.06} = 5437 \text{ lbs. per square foot;}$$

$$= 35.75 \text{ "   "   "   inch.}$$

The horse-power,

$$\frac{37.75 \times 200 \times 4 \times 28 \times 2}{33000} = 51+.$$

The relatively small capacity for its volume of cylinder and the relatively low mean pressure for the high initial are features to be observed. The engine referred to has long been regarded as a classic, but it was abandoned from the difficulty of maintaining the heating-chamber. Usually it was run at a lower temperature and pressure, and developed an average of 20 H.P.

Laubereau's engine is a more modern form of the Stirling.

271. Hot-air Engine with Temperature Changes at Constant Pressure. Ericsson's Engine.—The hot-air engine of 1852–3 designed for a 2200-ton sea-going vessel by John Ericsson has much the same classic and historic interest as the Stirling engine. The engine was calculated to be of 600 H.P., but actually ran at about 300. There were four cylinders, each 14 feet in diameter and having 6 feet stroke, causing

nine revolutions per minute. The air entered at about 380° F.
It burned 8 tons of coal per twenty-four hours, which is 1.1
pounds per H.P. per hour if the larger figure is taken, and 2.2
pounds if the smaller figure is correct. Its bulk and weight
were its defects. In 1854 it was replaced by a steam-engine.
In 1860 Ericsson brought out his early design of horizontal
small motor, and later (1880) the latest form adapted for
house-pumping was produced. Fig. 129 shows the Ericsson
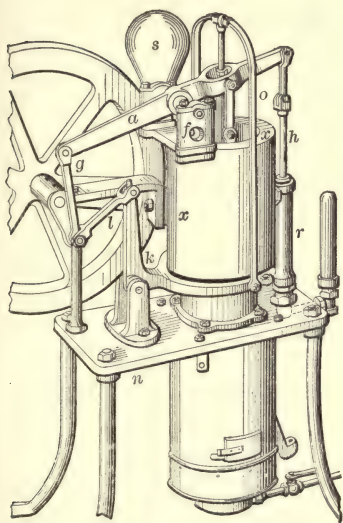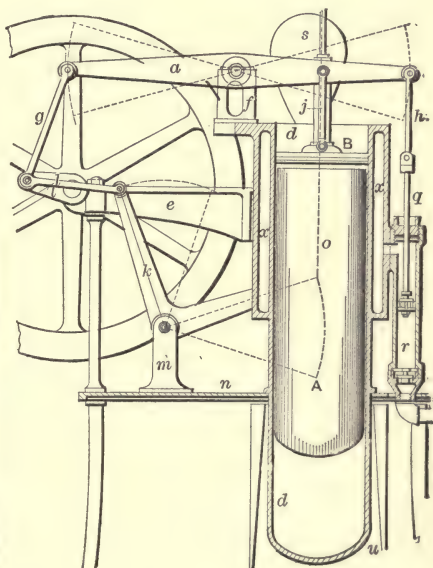
Fig. 129.   Fig. 130.

pumper in perspective, and Fig. 130 in section. The fire of
coal or gas is below the longer cylinder *d*, which again is water-
jacketed at the upper end, *xx*. In tank-pumping engines the
pumped water circulates through the jacket. *A* is the hollow
displacing piston, and *B* is the working piston proper. The
displacer is coupled to the bell-crank beam *K*, and so to the
crank, while the beam proper is linked directly to the crank.

Hence the two pistons are practically quartering. Fig. 131 shows a reproduction of an actual pv diagram, and Fig. 132 the theoretical curves. The straightness of the isothermal

FIG. 131.
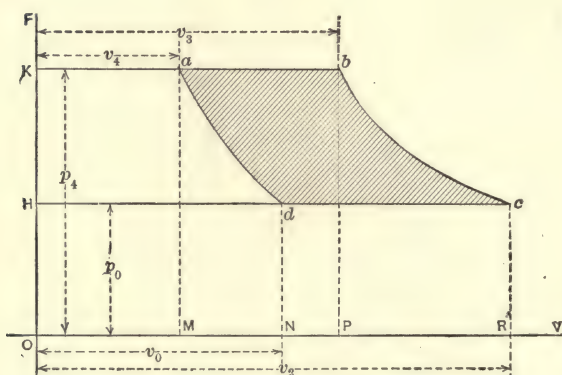
FIG. 132.

lines shows the variation of pressure to be slight compared with the change of volume.

The temperature-entropy curve diagram for the Ericsson engine will be the same as for the Stirling (Fig. 128, § 270), except that the coefficient will be C_p for the logarithmic curves, instead of C_v.

The Rider hot-air pumping-engine separates the hot and cold cylinders, and places a regenerator H between. The Ericsson pumper does not use a regenerator (Fig. 133).

272. Other Forms of Hot-Air Engine. — The engineers and designers of the continent of Europe have made more trials

with hot-air engines than have been made in America, but the limited extent to which even successful forms have been introduced have made their engines but little more than names. Richard Unger combined a separate compressing cylinder for
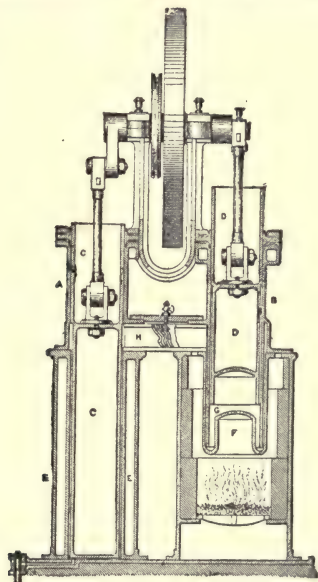
FIG. 133.

raising initial tension, with the plan of lowering the initial volume of such air by injecting cool water into it. Some of this water heated by the compression becomes steam. The hot products of combustion from the furnace mix with part of the cooled compressed air, and thence go to the valve-chest of the engine.

The Stirling-Laubereau engine had the working piston acted on at high heats by the air used, and a certain volume of air fills clearance volumes which underwent heat changes without doing work by its expansion. Lehmann's hot-air engine combined the low piston temperature secured by Ericsson, and

avoids unnecessary heating and cooling of a part of the air at
each revolution,

The plan of separate compression and the use of high-
tension air in a closed cycle are features of the Woodbury-
Merrill engine of American origin.   Other American designers
have been Shaw, Roper, Wilcox.   Other foreign types are
represented by Franchot, Siemens, Robinson, Bailey.

Special interest, however, attaches to a principle put forth
as early as 1851 by Dr. Joule, which leads to the operation of
internal-combustion engines, to be later discussed.

**273. Hot-air Engines with Separate Compressing
Cylinder.**—It is necessary to add to the organs of a hot-air
cycle-engine a pump which shall draw in air from the atmos--
pheric supply without, and deliver it to the working cylinder
if the cycle is to be an open one.   This aspirating pump is
usually also a compressing pump, driven from the working
shaft, and hence absorbing a certain amount of the energy
developed by the heating of the air.   The energy required for
compression is usually restored completely by the increased
energy of the working stroke, but of course extra weight is
needed in the fly-wheel to redistribute this restored energy.

This separate pump is a prime requisite of products of
combustion engines, and is present also in others.

The ideal indicator-diagram of such an engine on the pv
plane will have a form such as Fig. 134.

The pump-piston starting at the point represented by d
above the vacuum line a distance corresponding to atmospheric
pressure draws in a volume $V_0 = Hd$, representing that of one
pound.   By the return or compressing stroke of the pump the
air is first compressed adiabatically along da, and when the
pressure of the receiving chamber (the furnace of a products-of-
combustion engine) is reached, the valves to it open and the
compressed charge enters.   The pressure is practically constant
because the working cylinder is withdrawing air during this

displacement process, provided the heating chamber is of suf-
ficient volume, but the heat is not supplied isothermally as in
previous cases.

The working cylinder draws hot air at constant pressure
from the receiver or furnace to an extent represented by
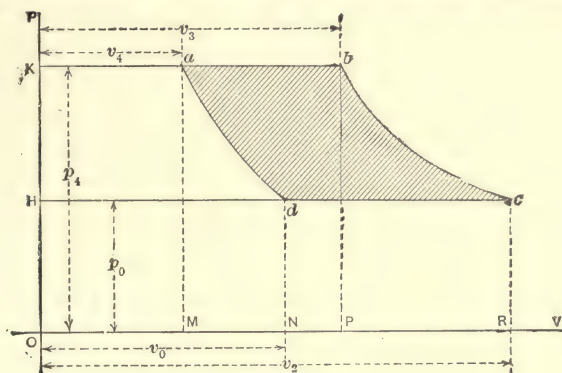
Fig. 134.

the line Kb, when cut-off occurs and adiabatic expansion down
the line bc, until atmospheric pressure is reached if the expan-
sion is complete, when the exhaust opens to the atmosphere
and the line cd is the return stroke. Such an engine is like
the combination of air-compressor and air-engine, or steam-
engine without condensation, whose diagram is closed by the
compression of the boiler-feed pump. The net work is the
shaded area, the white or plain part being the work of the com-
pressing pump. Using as subscripts the notation of Fig 134,
the area of the feed-pump work diagram will be $KadH$, which
will be the sum of

$$NdaM + KaMO - HdNO.$$
$$a = C_v(T_4 - T_0) + P_4 V_4 - P_0 V_0$$
$$= C_v(T_4 - T_0) + R(T_4 - T_0)$$
$$= C_v(T_4 - T_0) + (C_p - C_v)(T_4 - T_0)$$
$$= C_p(T_4 - T_0).$$

The area for the working-cylinder pv diagram will be similarly found to be $KbcH$, if the same scale of volumes be used as for the pump, which will become

$$A = C_p(T_2 - T_3).$$

The difference, or net work, will be

$$A - a = C_p(T_1 - T_2 - T_4 + T_0).$$

The heat expended in the receiver is expended in raising the temperature of the air from that caused by the adiabatic compression T_4 to the temperature T_2. Calling it Q, as the pressure is constant,

$$Q = C_p (T_1 - T_4).$$

T_1 is known as the receiver temperature, and T_4 can be calculated when T_0 is atmospheric temperature by the adiabatic formulæ (§ 154).

The heat rejected is given out to the atmosphere during any cooling from T_2 (at which the air exhausts) down to T_0, the atmospheric temperature. T_2 is similarly found from the T_1 value by the adiabatic formula (§ 154). Hence the efficiency is

$$E = \frac{A-a}{Q} = \frac{T_1 - T_2 - T_4 + T_0}{T_1 - T_4} = 1 - \frac{T_2 - T_0}{T_1 - T_4}.$$

Since the volume of the feed-pump should be to the volume of the working cylinder in the relation of the temperatures at the points a and b, the pressures being constant, and these volumes should bear also the same relation at d and at c, the equality can be written

$$\frac{V}{v} = m = \frac{T_1}{T_4} = \frac{T_2}{T_0},$$

by which relations the equation of efficiency transforms into

$$E = 1 - \frac{T_2 - T_0}{T_1 - T_4} = \frac{T_1 - T_2}{T_1} = \frac{T_4 - T_0}{T_4}$$

when working with complete expansion. The latter expression shows the value of increased compression by the feed-pump, and of high pressure in the receiver to which the T_1 corresponds. The first expression shows the value of having the expansion go down to the temperature T, if possible, when the efficiency would become unity, as would be expected. This means, however, that the compressing pump must be more nearly equal to the working-cylinder volume than is convenient or practicable.

274. Temperature-entropy Diagram of a Hot-air Engine Changing Temperatures Non-isothermally.—The Stirling and Ericsson engines changed temperatures at constant temperatures; the foregoing type changes temperatures at constant pressure, but the temperature is changing during the heating process. If, therefore, the pv diagram be as shown in Fig. 132 or 134, the te diagram will appear as in Fig 135. The
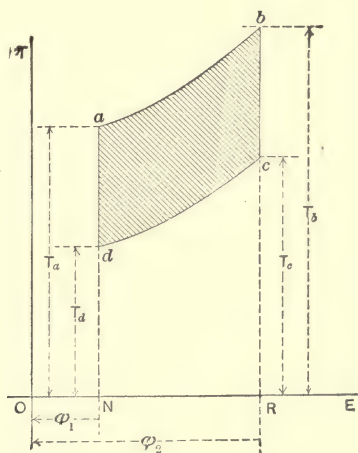
Fig. 135.

adiabatic change from d to a from the compression of the feed-pump is followed by the logarithmic change of entropy caused

by the heating process; such change, being at constant pressure, will produce a curve whose abscissa at any point will be

$$\phi = C_p \, (\text{hyp. log } T - \text{hyp. log } T_a)$$

when the temperature T is given. Then from b to c will be the adiabatic expansion in the working cylinder, and c to d the cooling under constant volume during the exhaust process back to the point in the temperature range where compression begins by the pump. The ratio $\dfrac{Ta}{Tb}$, which is $\dfrac{Na}{Rb}$, should be the same as the ratio $\dfrac{Ka}{Kb}$ on the pv diagram, and is the ratio m between the pump and working cylinder which it supplies The efficiency will be

$$\frac{\text{area } abcd}{\text{area } abRN} = \frac{\phi(T_a - T_d)}{\phi T_2} = \frac{\phi(T_b - T_c)}{T_b},$$

which is less than the perfect engine would offer which took in and rejected all the heat at the same extreme limits of temperature. In proportion as the curves ab and cd approach straight lines, by as much does the area of the mechanically utilized work approach the area of the perfect engine diagram.

275. Joule's Equivalent Hot-air Engine with Closed Cycle.—While the foregoing designs are of present practical interest, it may be desirable to say that Dr. Joule proposed a closed-cycle engine, involving the same transformations in 1851. While the engine was never built, yet its reverse lies at the basis of certain types of refrigerating machine, and its cycle would be the equivalent closed cycle to that of an actual products-of-combustion engine. Fig. 136 shows a diagrammatic scheme of such a closed-cycle engine. C is the piston of the compressing-pump, on the same rod with M, the piston in the working cylinder. H is the furnace-chamber at T_1, and C is the receiver with cooling water circulating in its tubes to maintain it at T_2. If these were large enough, the pressures in

them would not vary. The air compressed by C to T_a passes
through the valve v and is further heated to T_1, and expands
through u to drive M. On expanding into the cool-chamber
C the temperature drops to T_c by adiabatic expansion, and
from T_c to T_d by logarithmic cooling, as above explained.
The only changes necessary to transform Joule's engine into
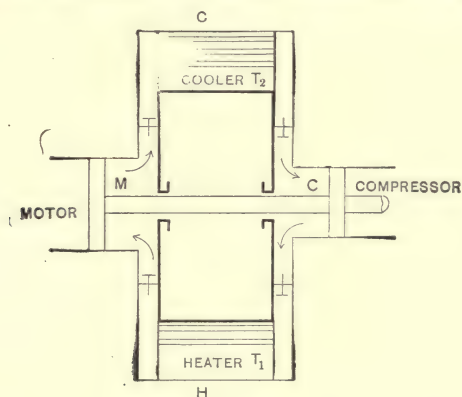
FIG. 136.

an internal-combustion engine in principle are those involved
in making the chamber H the furnace-chamber. with provisions
for introducing the fuel into it as required. The chamber C
can also be the outside air into which the working cylinder
exhausts and from which C shall take in its supply at T_0 at
each stroke.

276. **Internal Combustion Hot-air Engine Using Solid
Fuel.**—The introduction of the gas-engine and the oil-engine,
and the perfecting of the processes for gasifying fuel in pro-
ducers (Chapter V), have resulted in giving to the engines of
the earlier inventors an interest which is merely historic, in
their attempts to heat the working air by passing it through fire
of solid fuel (Fig. 137). The furnace was placed in a chamber
strong enough to withstand the pressure p_1. The compressing-
pump B forced air below the ash-pit up through the fire, where

it was expanded by heat and by combination with carbon. Being admitted to the working cylinder, against the piston A, it was exhausted into the chimney-stack. The furnace had to be charged with fresh fuel through a combination of double-
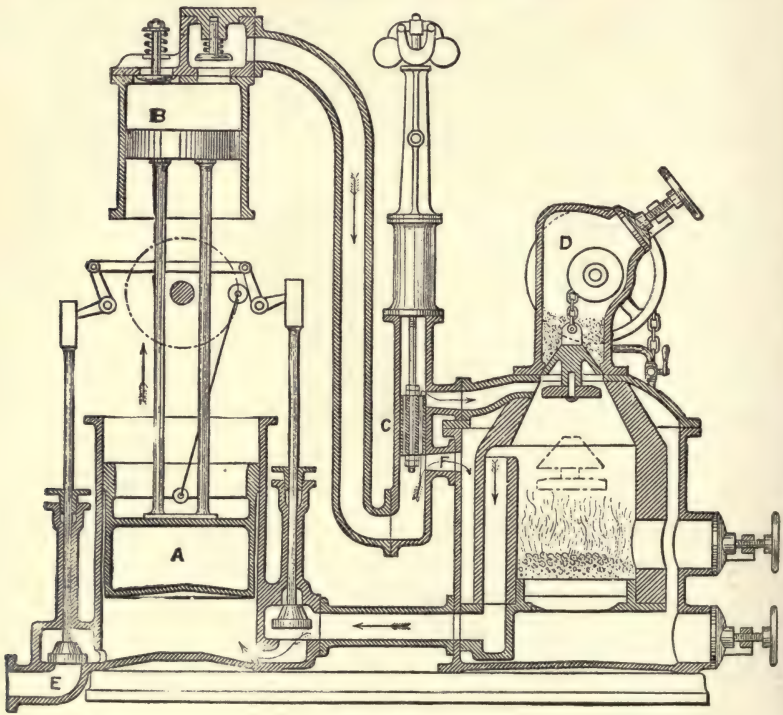
FIG. 137.

doors on the air-lock principle, D. The foreign names identified with this type of engine are those of Sir George Cayley and of Wenham (1873) and Duckett in England, and Dr. Avenier de la Gree in France. The American engines have been those of Shaw, Roper, and Wilcox. The difficulties have been those caused by flue-dust and grit in the cylinders; the rapid destruction of working surfaces and valves by the intense heat, and the difficulties of lubrication. They were also more bulky

than the closed-cycle engine in proportion to the power developed. If it were desirable to follow their working, the foregoing principles can be applied, using for initial volume not that of one pound of air, but the volume produced by supplying one pound of air to the furnace, as given by the data of Chapter IV.

277. Concluding Summary.—The hot-air engine in small sizes is more economical than the steam-engine of the same capacity. In larger sizes it has about the same economy as the less economical steam-engine, measured in coal consumed per horse-power. It has the advantage of avoiding the steam-boiler as a magazine or reservoir of energy which may be liberated by accident so suddenly as to be explosive. It can be run by less skilled and expensive labor and no steam-runner's license is demanded. It is safe and odorless.

The objections to the hot-air engine are the greater bulk and greater weight for the same power than is required with the steam-engine; the low mean pressure with high initial pressure, which latter compels great strength of structure; the deterioration of heating-surfaces exposed to high heats and consequent oxidation; the difficulties of packing and lubricating at high temperatures; the difficulty of regulation closely to varying resistance.

If there is any danger to the present supremacy of the steam-engine, it will be in relatively small plants that a hot-air engine can be a substitute; the gas or internal-combustion engine is more to be feared than the hot-air engine proper.

CHAPTER XX.

INTERNAL-COMBUSTION ENGINES.

THE GAS AND THE OIL ENGINE.

280. Introductory—Historical.—The first suggestion of an engine exploding a mixture of gas and air behind a piston belongs to a considerable antiquity, when gunpowder was also similarly considered by Huyghens as early as 1680. The first English patent dates back to 1794 (Robert Street). The Lenoir engine of Paris in 1860; the Hugon engine of 1865, with water injection; the Otto and Langen (Cologne, 1867), while great advances upon the early types are only of historic interest since the introduction in 1876 of the "Otto Silent" gas-engine and the cycle represented by this class. The idea of compressing the gas-charge before explosion was brought forward as early as 1801; early names in English practice are those of Barnett (1838) and Sir C. W. Siemens (1862). French engineers have been Million (1861) and Beau de Rochas (1862), who proposed the four-stroke cycle, now known by Dr. Otto's name. The Dugald Clerk engine (1880) and the Atkinson (1885) are types involving features not common to all which have been recently brought forward under a great variety of trade or proprietary names.

It will be seen that the introduction of a combustible gas into a mass of air required to burn it, and the ignition of the gas in the air within a confined volume, results in an expansion of that air which will exert a pv pressure, which can be made to do work by means of piston and engine mechanism. The

gas engine is therefore a hot-air engine of the internal-furnace class and works upon an open cycle, since it draws in a fresh charge of gas-fuel and air at each working stroke, and rejects the products of the combustion with the exhaust. It is also a " products of combustion " engine, but using a fuel without grit or ash. Oil-engines, gasefying the oil and mixing it with air, belong to this same class, requiring only proper provisions for pumping and vaporizing the oil-fuel.

The fuel-engine cycle (§§ 273–75) is the usual and typical
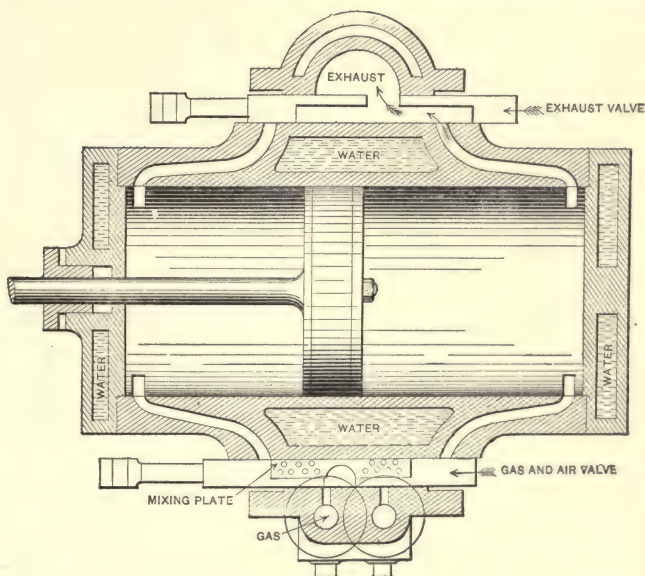
Fig. 140a

hot-air gas-engine cycle. Chemical considerations must be borne in mind for their effect in modifying both the theory and action of the gas-engine.

281. Lenoir Gas-engine of 1860.—The cylinder of this early engine was a water-jacketed steam-engine cylinder (Fig. 140). During the first part of the outgoing stroke air and gas

in proper proportions of 1 of gas to about 12 or 14 of air air were drawn into the cylinder by aspiration. The inlets being closed, the mixture was fired by a spark from a
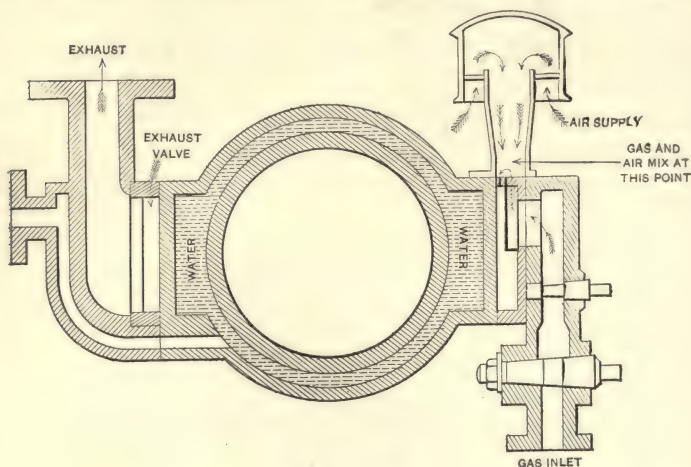
Fig. 140*b*.

Ruhmkorff induction-coil. The increase of volume following the explosive ignition increased the pressure at about half stroke, which fell till the end was reached, when the expulsion of the products of combustion took place on the return of the piston. Fig. 141 shows a typical *pv* diagram. It took 95
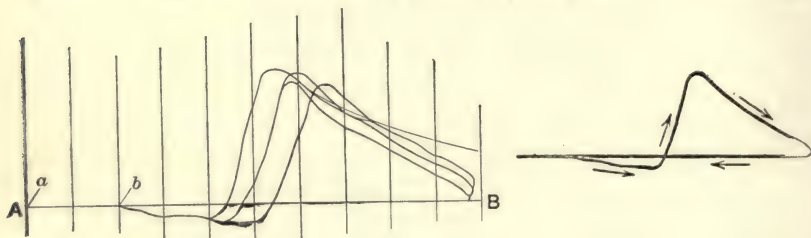
Fig. 141.

cubic feet of gas per horse-power per hour (which is more than four times the present requirement), and the high temperature and the noise were objectionable. The platinum points of the

sparking electrodes were also liable to become clogged with lampblack or soot deposit. A large amount of the available power was lost in the impact effect, whose heat was absorbed and wasted by the jacket.

282. Hugon's Gas-engine of 1865. — Hugon's improvement on the Lenoir type followed from the injection of water with the gas and air mixture. The vaporization of this injection absorbed some heat and gave it out in the work of expanding, although the expansive force was diminished. The distribution of the effort made the moving parts suffer less and diminished repair expenses, and made the engine more durable, especially at packed surfaces, such as piston surfaces, stuffing-boxes, etc. Gas-jets were used to fire the charge.

A form of engine known as Bischoff's is the only survivor of the purely explosive non-compression type, and burns 120 cubic feet of gas per H.P. per hour.

283. Otto and Langen Atmospheric or Free - piston Gas-engine of 1867. — To avoid the loss from impact effect, when positive connection to the crank-shaft forced the piston to yield but gradually to the sudden increase of volume of the charge, Otto and Langen, in Germany, and Barsanti and Matteucci, in Italy, proposed to have the vertical cylinder of some height, and the piston-rod to rise without effect on the crank-shaft during the stroke caused by the gas ignition. The piston yielded like a projectile and rose to the top of its traverse. The sudden expansion of the gas-mixture cools it also suddenly, and as its tension falls below the atmospheric tension, the pressure of the atmosphere acts to force the piston back downward. The piston-rod is connected to the shaft by a rack and pinion-gear, operated by a pawl and ratchet-wheel, so that the pawl clicks idly on the up-stroke, but the rack and pinion transmit the atmospheric effect on the down-stroke (Fig. 142). This engine was obviously noisy and irregular. It consumed, however, about 30 to 40 cubic feet of gas per H.P.
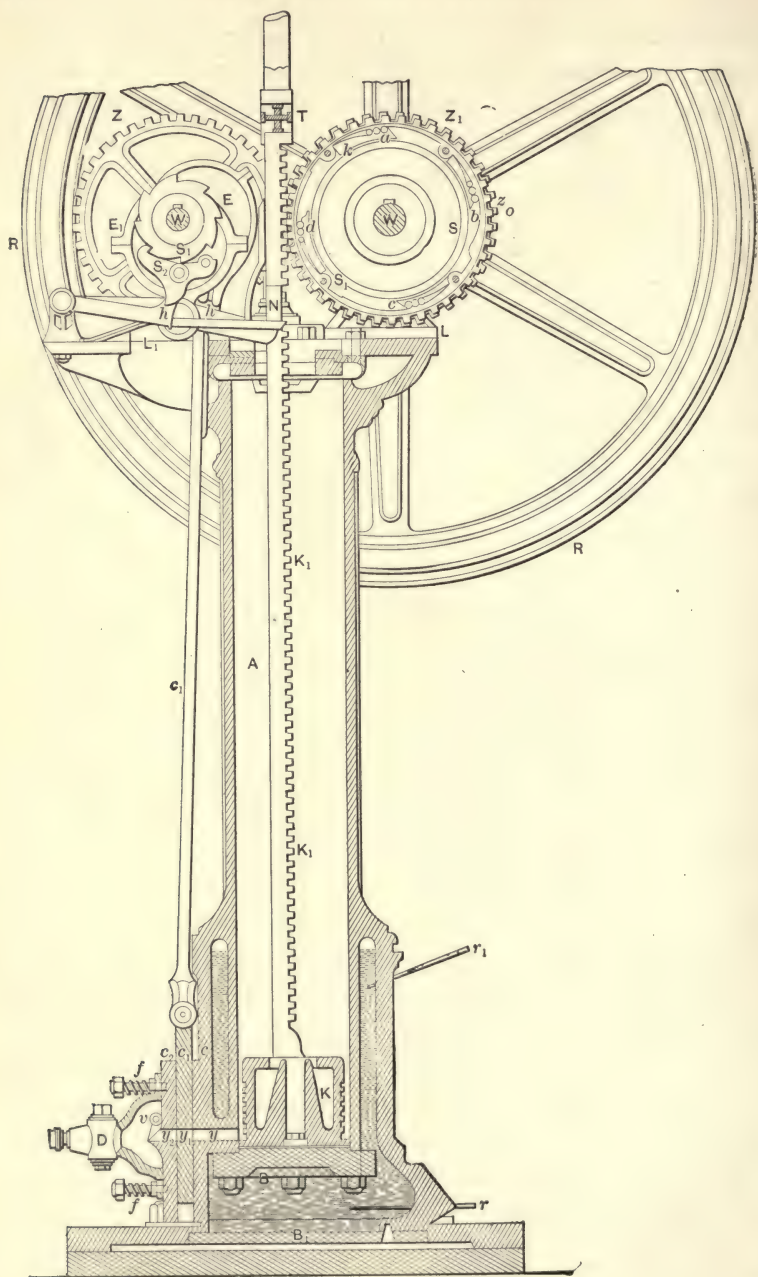
FIG. 142.

hour, and was manifestly a distinct gain on its predecessors. Ignition was effected by an outside gas-flame.

284. Brayton Gas-engine or Ready Motor of 1873.—A gas-engine forming a type by itself was brought out and introduced to a limited extent in eastern America in which a mixture of gas and air drawn into a pump-cylinder on its aspirating stroke was compressed by the return stroke into a reservoir under a pressure of about 70 pounds per square inch. From this reservoir it was allowed to flow to the working cylinder during perhaps one-half of the outgoing stroke of the piston. The mixture was ignited in the working cylinder by a flame, the back-flow of flame to the reservoir being prevented by wire-gauze at the inlet. The mixture thus simply increased in volume but without rise in pressure, since the connection with the reservoir was still open, and followed the piston up to cut-off at half stroke. The rest of the stroke was operated by the expansive energy of the gas-mixture, and on the return stroke the contents of the cylinder were exhausted. Such an engine had no explosive ignition, but the slow inflammation took place as fast as the mixture was admitted and was continuous during such admission. The pump-piston had the same cross-section as the working cylinder in the beam form of engine, but one-half the stroke. In steeple-engines the two pistons had the same stroke, but half the area was given to the compressing piston.

The principle of slow inflammation is wasteful with a water-jacketed cylinder, since a loss of energy or increased gas-consumption results from the necessity of sustaining pressure and temperature. The terminal or exhaust temperature of the gases was high from this cause, as well as the mean temperature. The efficiency as measured by the work done by 1 cubic foot of air with the Brayton engine was, however, 0.36, as compared with 0.21 given by the previous types, having no compression and working by explosion. It was displaced, however, by the superior economy of the Beau de Rochas or Otto

four-phase cycle, whose efficiency was measured by 0.45 for a cubic foot of air used under the same assumed conditions.

285. Four-phase Cycle of Beau de Rochas.—What is generally called the Otto cycle was first suggested in a French patent of 1862 by Alphonse Beau de Rochas, who also advocated the advantages of previous compression of the combustible mixture, and proposed to do away with the separate compressing pump by making only one stroke in four to be the working stroke in a single-acting engine. The Beau de Rochas or Otto cycle involves:

1. Aspiration of the mixture of gas and air in proper proportions during an out-going stroke of the piston (1–2 in Fig. 143).

2. Compression of the mixture by the return of the piston (2–3–4). This compression fills a comparatively large clearance volume behind the piston, which must be so adjusted to the displacement by the piston that there shall be no danger of such elevation of temperature from the compression as to ignite the mixture as the result of compression alone.

3. The piston being at or near its inner dead point (4), the compressed mixture is ignited by some acceptable and reliable device, at which the pressure rises at once (4–5) and exerts its outward effort to drive the piston forward. Expansion is followed by gradual lowering of pressure during this working stroke (5–6–7).

4. The products of combustion are discharged into the open air through the exhaust-valve by the return of the piston to its inner dead centre (8–1). The cycle then repeats itself.

It is apparent that a heavy fly-wheel must be used to equalize the motion of the crank-shaft, having energy enough stored in it by the working stroke to overcome the resistance during the time of the other three strokes, and cause also the piston to perform the acts of the cycle in the cylinder. High rotative speed is therefore an advantage. Furthermore, a

high initial pressure and temperature are desired, with a low terminal value for both, so as to secure a high mean value. Rapid inflammation is therefore desired, and the methods of ignition become important.
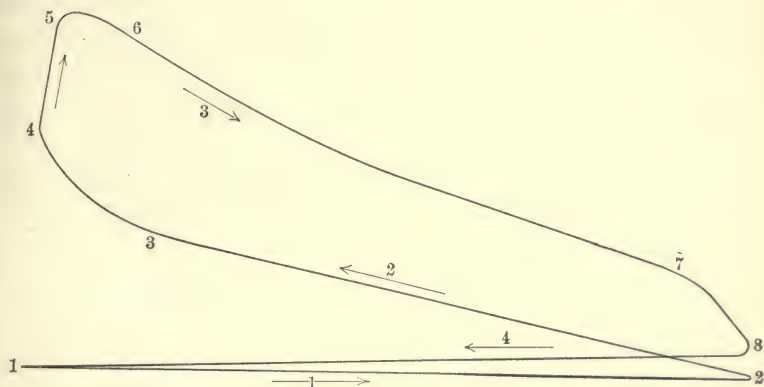
FIG. 143.
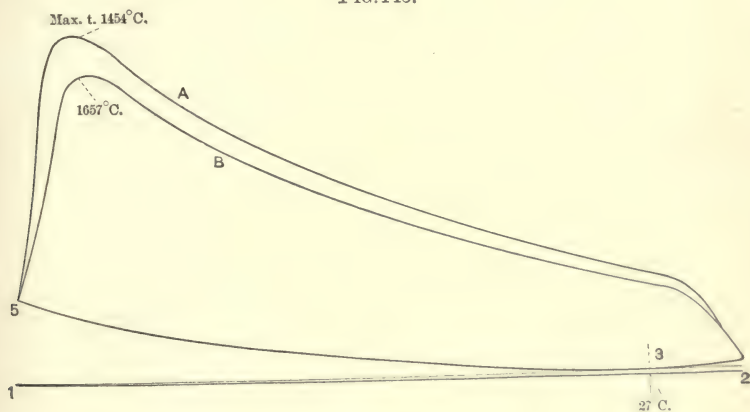
FIG. 143.

286. Otto Silent Gas-engine of 1876.—The propositions of the French patent were not embodied in industrial form until Dr. Otto reinvented the cycle and, as the result of much experimental study, brought out the prototype of all the

modern gas-engines, to which he gave the trade-name of "Silent," to distinguish it from his earlier noisy type.

The engine was single-acting, of trunk design to secure compactness. The cylinder and valve-casing were water-jacketed. The valve-gear was operated by a shaft driven from the main-engine shaft at half its rate. Cams opened the valves at proper intervals. Regulation of effort was caused by making the inlet-cam miss its stroke when a centrifugal governor driven from the main shaft was turning too fast. Hence this type of engine has become known popularly as the "hit-or-miss" gas-engine. The mixture was ignited in early forms by a flame. The clearance volume was about one half the cylinder volume. (Fig. 144.)
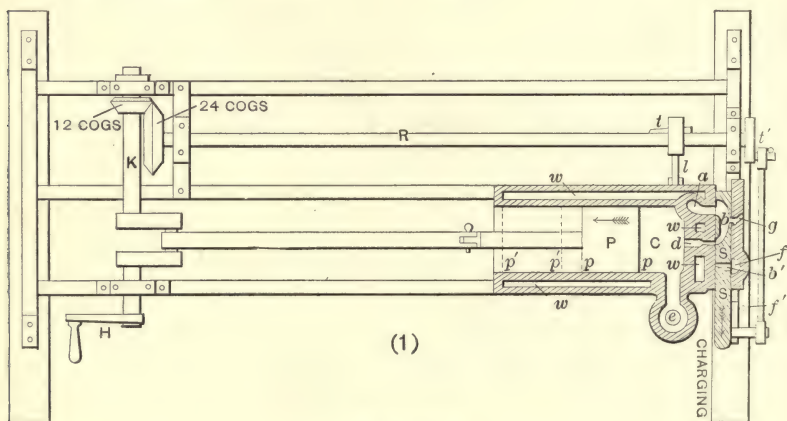
FIG. 144.

Under the Otto patents many modifications were introduced, such as to use two cylinders driving the same crankshaft, one of which should be two phases ahead of the other, and thus diminish the interval between working strokes. The cylinders have been placed vertically instead of horizontally; different igniting devices have been used; the front or idle side of the piston has been used as the aspirating and compressing side; double-acting types have been used with ignitions on both sides, occurring alternately. Since the

expiration of the Otto basal patents, many new forms and names have come forward, both in England and America.

287. Dugald Clerk Gas-engine of 1880.—The twinning of the cylinders to secure an impulse every revolution makes a costly engine by reason of the expense of the two mechanisms. The most satisfactory plan is to separate the aspirating and
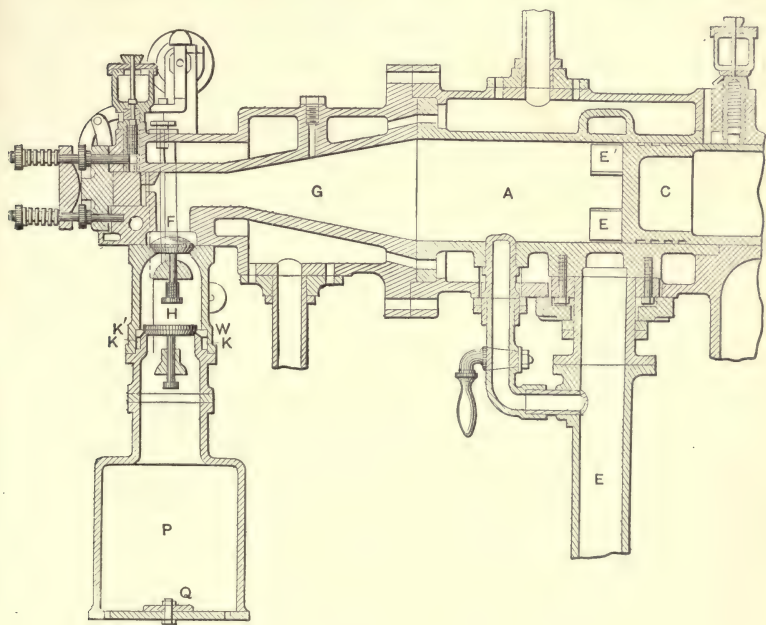
FIG. 145.

compressing operations from the other two doing the former work in the separate pump-cylinder. In the Clerk engine this arrangement is secured. At the back end of the working cylinder is placed the clearance volume, which is a conical or trumpet-shaped space, communicating through a lift-valve with the compressing cylinder. The pump or displacer crank leads the working crank by 90°. The exhaust-ports from the working cylinder are at the front or outer end of the bore, and are

uncovered by the working piston as it nears the outer end of its traverse. The displacing piston is then returning, compressing the charge aspirated on its outer stroke, and the flow of fresh mixture into the clearance drives the expanded products of combustion in front of it, expelling the previous charge through the exhaust-ports. The expanding trumpet-shape by lowering the velocity of the new charge lessens the likelihood of wasteful mixture of the new and old mixtures. Any union which does take place is a gain from preheating the new charge, and thereby cooling the old. The return of the motor-piston compresses the new mixture, which is ignited by a flame as the crank passes its dead-centre at each revolution. Fig. 145 shows the Clerk engine in section, A being the working cylinder, C its piston, G the conical clearance volume, and E the exhaust-ports.

288. Atkinson Differential or Cycle Engine of 1885.— The peculiar features of this engine are the unusual linkage between the pistons and the crank-shaft, whereby the pistons can be made to act as their own valves, and whereby a greater expansion is attainable than with fixed clearance volumes to be filled with gas-mixture. The engine has appeared in two forms. In that selected in Figs. 146 and 147 there are two trunk-pistons, linked to the crank-pin by the two curved beams The latter are borne on the two massive beam-centres, whose location to each other is so chosen as to force the pistons out of symmetry or phase as the crank passes the four cardinal points as indicated in Fig. 147. In the first position, with the crank at the extreme left, the pistons are close together. An automatic lift-valve admits a charge of gas and air between them as they separate by the quarter revolution of the crank to the position 2. This movement, and the further movement toward position 3, closes the admission- and the exhaust-port, and as the pistons move toward each other, compression of the charge takes place. The compression being completed in

position 3, the charge is fired and the pistons separate, performing the working stroke, expanding the mixture by the increase of volume. The exhaust-port is uncovered and discharge begins when the pistons reach position 4, at which time the left-hand piston is moving rapidly to the right, while the left one is nearly stationary. The four usual operations of

ELEVATION
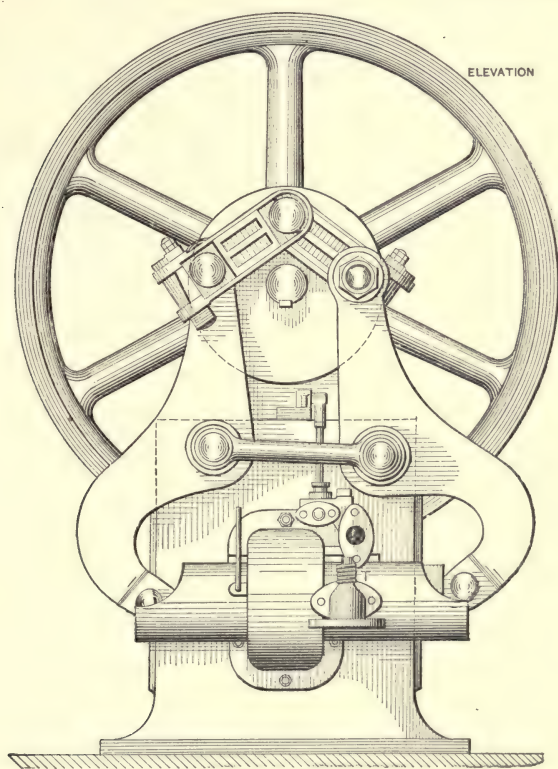
Fig. 146a.

the cycle are provided for in the single cylinder, and an impulse occurs under full load at every revolution. The unusual character of the Atkinson linkage limited it to experimental sizes and low speeds and powers, and the design modified from it never attained any commercial importance for business reasons.

PLAN
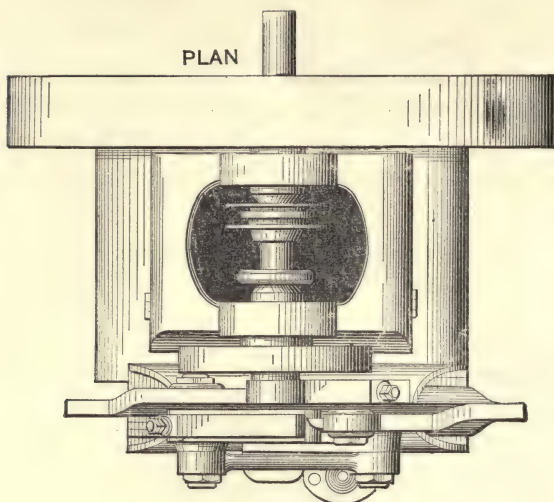
FIG. 146*b*.
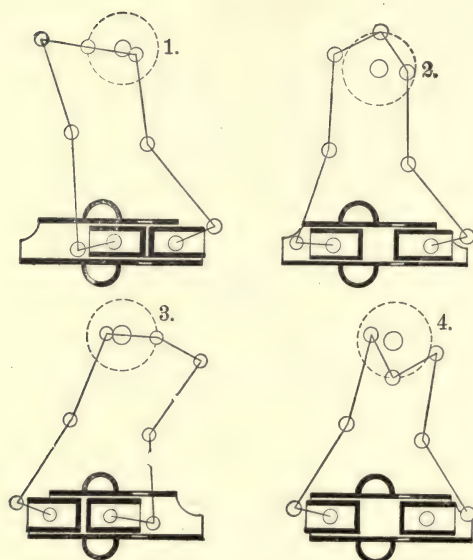
FIG. 147

Fig. 148 shows the cycle or single-piston engine of Atkinson. The connecting-rod does not drive the crank directly, but
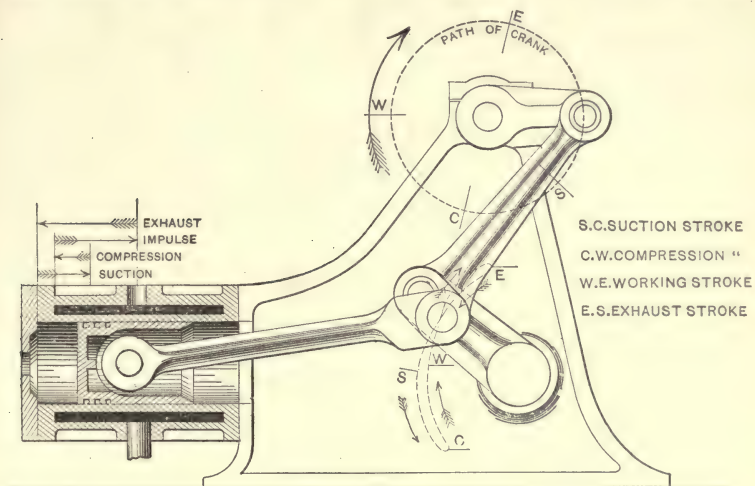
FIG. 148.

a toggle-lever, pivoted at the centre as appears in Fig. 149. The connecting rod to the crank is T shaped, bearing at its lower
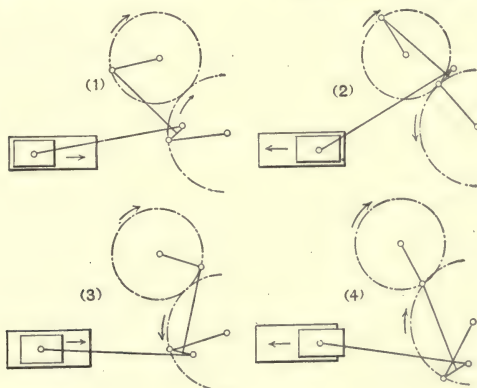
FIG. 149.

end the short transverse portion which forms the arms of the T. The piston-rod couples to the further end of this trans-

verse member, and the toggle-lever is jointed to it at the end. It will appear from this ingenious arrangement that the piston will make two strokes in each direction for one revolution of the crank, and that these will not be of the same length. The first outstroke or intake stroke will be longer than the return or compressing stroke, so that a compression volume is left. The explosion then occurs followed by the longest outstroke, giving the greatest expansion volume. Then the longest stroke of all takes place, from furthest out to furthest in, sweeping out all products of combustion. The toggle-lever causes the doubling of the strokes, since it swings through an arc only, and must do this twice in one revolution. The position of the cross-arm of the T as the toggle-lever swings is what gives varying length to the piston traverse.

289. Classification of Gas-engines.—The foregoing paragraphs have made it apparent, therefore, that there are two general classes of gas-engine.

Class I. Those making no use of compression;

Class II. Those employing compression.

The first class contains the purely explosive types and presents the two sub-classes :

(*a*) Explosion drives the power stroke ;

(*b*) Explosion lifts the piston freely, and the return is the working stroke.

Class II contains the modern efficient engines and may be divided into :

(*a*) Compression effected in working cylinder ;

(*b*) Compression effected in a separate or pump-cylinder.

They may again be grouped into two sections according to the working of the expanding heat medium :

(*c*) Ignition occurs at constant volume, followed by a sudden rise of pressure ;

(*d*) Ignition occurs at constant pressure from a reservoir ; the gas mixture burns slowly as volume increases.

The Otto engine belongs in groups (*a*) and (*c*), the Clerk engine in groups (*b*) and (*c*). The Brayton engine is the only example of group (*d*) and it is also in group (*b*).

290. Methods of Igniting the Gas-charge. — Gas-engines may again be grouped into four classes upon their mechanical side according as they vary in the method used to set fire to the mixture of gas and air which has been drawn into the cylinder by the aspirating stroke. The methods practised by different designers for this purpose may be grouped as follows :

(1) Electrical methods ;

(2) Flame methods ;

(3) Incandescence methods ;

(4) Compression of the charge.

The first plan is an early one. A pair of naked electrodes allow a spark or an arc to pass between them at the time when they are exposed to the gaseous mixture by the withdrawal of a slide, or the spark passes as a contact is broken between two poles exposed in the clearance. The breakage of continuity is timed by the motion of the valve-gear.

This plan avoids the inconvenience of opening the cylinder cavity by valve-movement when that cavity is under greater pressure than prevails without it, and a consequent tendency to leakage and unpleasant odor. The difficulties are those belonging to electric methods in unaccustomed hands, the clogging of the electrodes by lamp-black deposit, and the annoyance in finding the cause when ignition fails. The best electric igniters have the contacts wipe over each other so as to be kept cleansed by the scraping action.

The second plan is also an old one. It was used in the Barnett engine (1838) in a typical form (Fig. 155), depending on what may be called the air-lock principle. An external flame *A* set fire to a stream of gas entering the hollow cock chamber from below when the opening 1 was turned toward

such outer flame. The rotation of the cock through 90° clock-
wise in the plan brings the flame in contact with the explosion
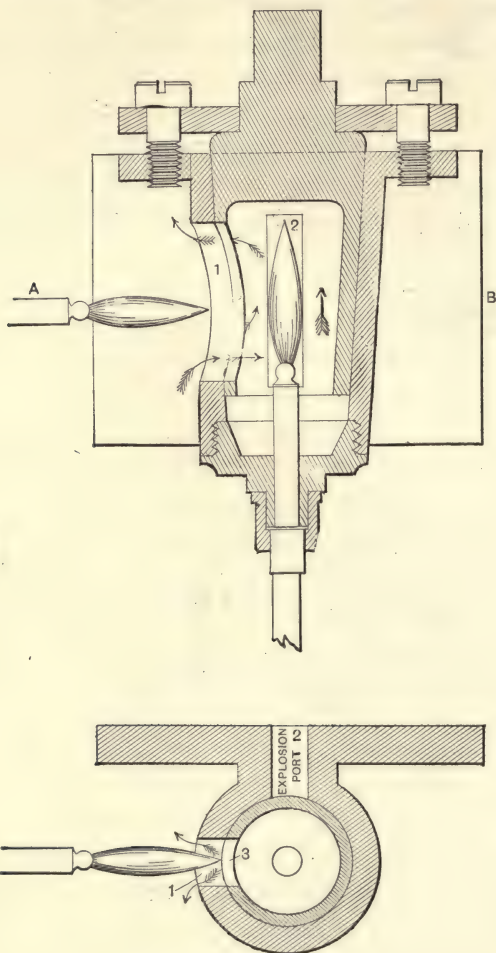
Fig. 155.

port 2 and fires the charge, blowing out the enclosed flame, of
course, in the process. It is relighted by the reverse motion
of the cock. The intermittent contact of the flame and the

explosion port can be secured by a sliding valve, which carries the igniting flame first to the exterior flame and then draws it back into line with the port. This plan requires the presence of sufficient air to ignite the gas-flame in the valve-chamber. A plan introduced by Clerk of using the normal mixture of gas and air for the igniting flame has much increased the rapidity at which ignitions can occur. In the Brayton engine the flame was continuous on the upper or inner side of the gauze through which the mixture streamed.

The third class of igniters uses either a short coil of incandescent wire kept hot by a current of electricity of sufficient potential running through it, whose incandescence can serve to ignite the mixture when a slide is withdrawn from in front of the coil or the latter is moved into line with the explosion port; or else the mixture is fired by passing part of it through a tube or grating, which is heated to incandescence by a separate flame or by the heat of the explosion of the previous strokes. The tube may be of wrought iron or of some ceramic material, or (as in Clerk's plan) a measured stream of gas at each stroke can be admitted to heat a cage of platinum plates, which cage, being borne by a slide, is presented white hot opposite the ignition port at the proper time by the motion of the slide.

The fourth plan is one particularly applicable to oil-vapor engines, but capable also of igniting with a gas-mixture. It depends on raising the temperature of the gas-mixture by compression in a non-conducting clearance-volume to such a temperature that chemical combination is possible at that pressure and temperature, and occurs without external igniting appliances. This is of course in the incandescence class, only the temperature of the clearance-volume does not have to be so high by reason of the heat in the mixture due to the compression. This principle is used in the Hornsby-Akroyd and the Diesel engine.

A class of igniting methods might be made of certain suggestions proposing the use of spongy platinum, or the use of spontaneously igniting compounds such as phosphoretted hydrogen. They have not been reduced to successful practice, however.

The objections to the flame methods are the danger from extinction of an exterior flame by draught or otherwise, and the odor and leakage which seem inseparable from it. The electrical methods are uncertain and troublesome.

291. Indicator Diagrams from the Gas-engine.—The foregoing types will give each his peculiar indicator diagram of effort on the *pv* plane. Figs. 156 and 157 show the non-compressive type of engine, the former a Lenoir and the latter an Otto and Langen.

Figs. 158 and 159 show the motor cylinder and compressing pump of the Brayton engine.

Figs. 160 and 161 show typical diagrams from Otto and Clerk engines.

The lower lines in Fig. 160 show the aspirating and exhaust lines of the first and fourth parts of the cycle. For a Clerk engine, with separate displacing cylinder, a diagram such as Fig. 162 will be taken from that separate cylinder.

The interest attaching to the first four is at present only historic. With respect to the latter or Otto cycle diagrams it is to be noticed that the pressure does not rise to its maximum instantly, or while the engine is at its dead-centre, nor is it dangerously great in amount. The strain on the engine mechanism is no more exacting than that which comes upon an engine using steam of high pressure. It must be observed, however, that there is no considerable duration of that maximum pressure, but the drop due to expansion sets in at once. In other words, there is no way of increasing the power of a gas-engine by any process analogous to following the piston during a considerable part of the stroke with full boiler pres-
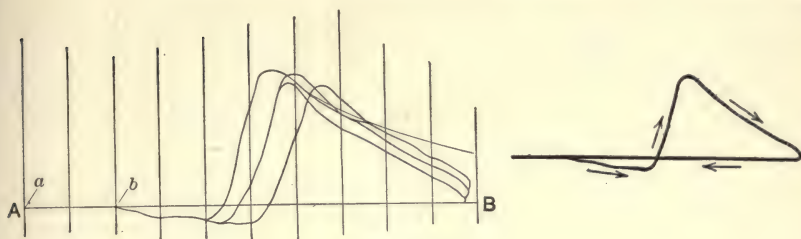
FIG. 156.
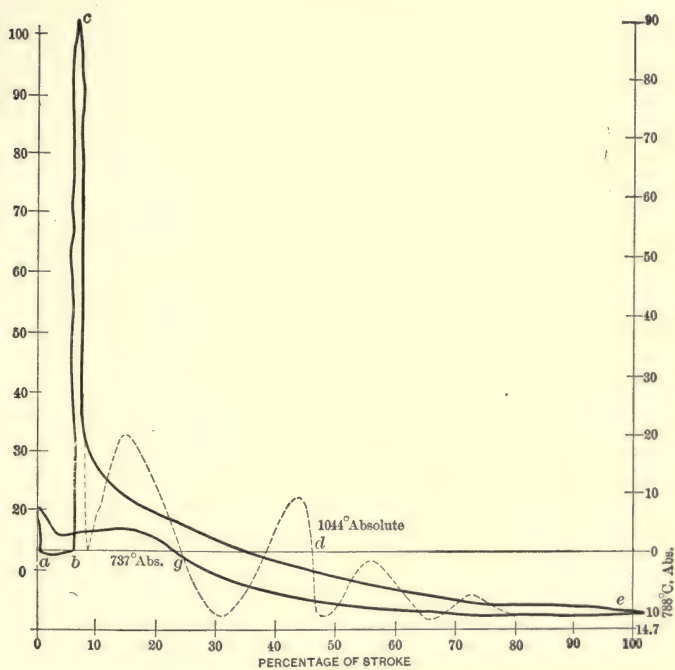
FIG. 157.
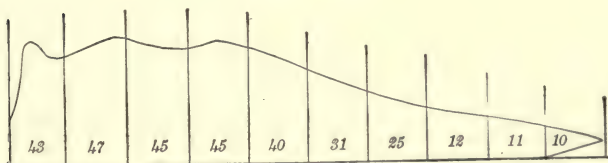
FIG. 158.
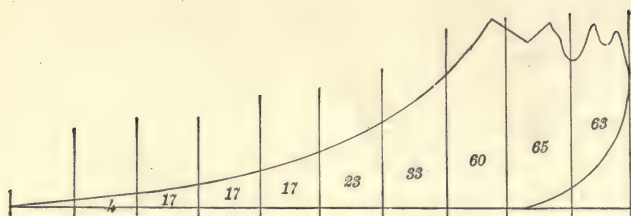
Fig. 159.
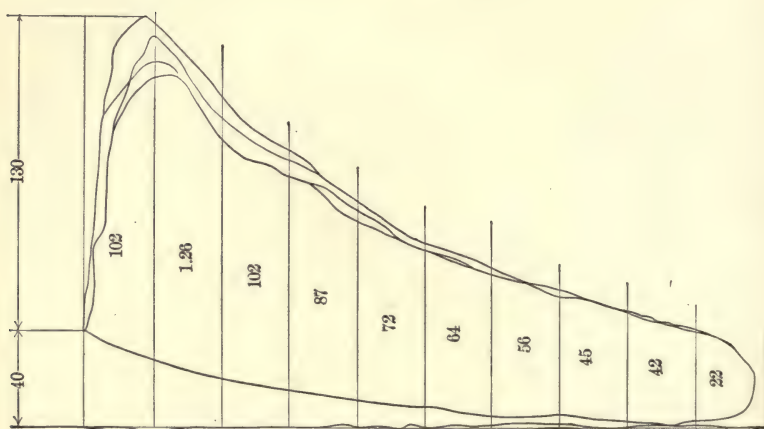
Fig. 160.
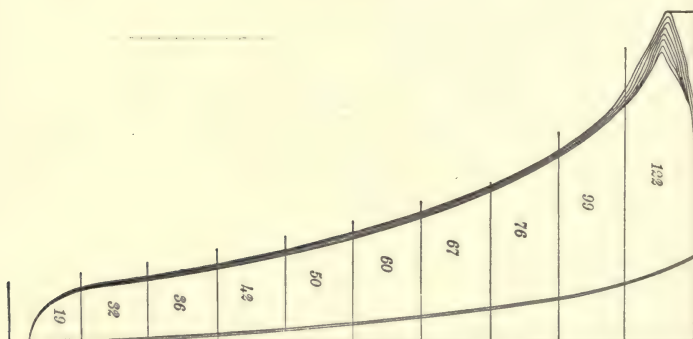
Fig. 161.

sure, which is possible with the steam-engine, if the valve-gear is constructed so as to allow it. It appears, further, that the quicker the complete inflammation of the mixture, or the less the increase of volume during ignition, the higher the initial pressure, and the greater the mean forward effort. It will be noted, furthermore, that it will be but rarely that expansion can be carried down to atmospheric pressure. It will usually be incomplete, entailing loss of available energy (§§ 201 and 255). The mean pressure must be such in the single-cylinder Otto engine as shall be proper for over four times the average horse-power, since the working stroke must not only do the external work of that stroke, but must also store in the fly-wheel an amount of energy for the external work of the next suceeding three traverses of the piston, and overcome the resistances in the engine itself in the compressing stroke.

FIG. 162.

Since the maximum pressures are caused by the flaming of the gas in the air, it becomes of interest to study the temperatures to be expected.

The questions, then, on which experimental knowledge must be sought will be the rapidity of flame propagation and the relation of pressures to actual temperatures.
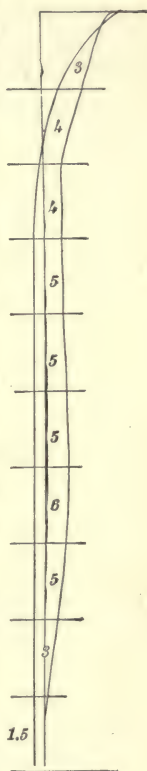
292. Some Phenomena of Ignition in the Gas-engine.— The volume of air required for the combustion of a gas of given composition has already been calculated (§§ 24–27), as well as the theoretical temperature of a flaming substance when its calorific power is known, the weight or volume and specific heat of the products of combustion being known or assumed (§§ 28, 61, and 68). In a gas ignition the weight of combustible per cubic foot is not large, but the time taken for its combustion

is very short.   Hence the reasoning of § 61 should be appli-
cable, and a high temperature should be secured.

The pressure behind the piston having been observed by
the indicator, the temperature corresponding can be calculated
from the Gay-Lussac law, as the volume is constant (§ 112).
The calculation need only be checked by the fact that there is
a slight contraction of volume resulting from the combination
of the combustible gas and air as two bodies into one chemical
mixture.   With ordinary street gas, and the minimum dilution
with air, this contraction will be less than four per cent, and
diminishes with the dilution.   Hence it is usually safe enough
to calculate temperatures without allowance for this contrac-
tion.

But complete combination does not occur at once, nor are
theoretical pressures and temperatures attained.   Physicists
appear to agree that nearly one half of the heat present in the
mixture of inflammable gas is kept back, and is prevented from
causing the increase in pressure to be expected from it.   Stan-
dard experiments made by Dugald Clerk have given the fol-
lowing results with English illuminating-gas (Oldham):

No.	Vol. Gas.	Vol. Air.	Pounds per sq. in. Gauge Pressure.	
			Observed Pressure.	Calculated Pressure.
1	1	14	40.0	89.5
2	1	13	51.5	96.0
3	1	12	60.0	103.0
4	1	11	61.0	112.0
5	1	9	78.0	134.0
6	1	7	87.0	168.0
7	1	6	90.0	192.0

The theories which have been advanced to explain this
suppression of heat are at least three.   Hirn's explanation was
that at these high temperatures the cylinder walls and water-
jacket absorbed heat so rapidly that at a certain point the
abstraction of heat was faster than its liberation by the burning

of the gas. The objection to this theory is that the suppression of heat does not seem to depend on the cylinder surface nor on the efficiency of the jacket. Bunsen's theory is that at high temperatures the phenomenon of chemical combination cannot occur readily, and that a high enough temperature may be reached to dissociate or break up combinations already made or making. Hence, if the combination cannot exist at any high temperature, the mixture will remain made up of the free constituents, which will come together only as the temperature falls. The fact of dissociation has been proved by physicists; its applicability to the gas-engine is still partly in question by reason of the fact that suppression of heat does not seem to increase in proportion as the temperature increases.

The third and most interesting theory is that advanced by Mallard and Le Chatellier, that at the higher temperatures the specific heats of the constituents and of the products of combustion increase. That is, if, as seems to be the case, the specific heats of oxygen and nitrogen double at or near 3600° F., the heat is completely evolved in the flame combustion, but twice as much heat is taken care of in heating these absorptive gases. This has the same difficulty as the dissociation theory; a greater proportion of heat should be evolved at the lower temperatures, which is not always the fact.

The soundness of the dissociation theory seems to be suggested by what is known as the "after-burning." The expansion should fall below the adiabatic line with a good water-jacket, since heat should be withdrawn and the pressure thereby lowered. The actual curve is nearly adiabatic or above it, suggesting an addition of heat during expansion, which of course must come from the combustion of gas in the charge which was not ignited at the beginning of the stroke. Such gas is not burned with the same economy as the explosion proportion, and liberates less heat per unit of weight. It fattens the indicator-card, however.

Dr. Otto's claim and theory for his silent engine was that the dilution of the fresh charge with the products of combustion of the previous one, and their arrangement in a species of stratification, caused a slower propagation of the flame, and moreover caused a cushioned effort of the expanded charge against the piston.   Later experiments have thrown considerable doubt over the validity of these contentions.

293. Usual Mixtures of Gas and Air.— The gas being the expensive element in the mixture of gas and air drawn into the cylinder, it would appear that to increase the proportion of air in a given cylinder volume was to increase the economy. This is only true, however, when the power of the cylinder is the fixed element, provided the diluted mixture does not lose pressure per square inch in a more rapid proportion.   That is, it may be necessary to increase the cylinder volume more rapidly to secure a given power, and so draw in more gas than if the proportion of gas were increased in a cylinder of given volume.   It is desirable, furthermore, that in a cooled cylinder the water in the jacket shall not lower the average temperature and pressure too rapidly.   Hence the most efficient mixture becomes a matter for experimental determination.   The mixture must further be one in which the flame propagation or the time of ignition of the mixture shall bear the desired relation to the speed of the engine, or the period of the single working stroke of the four-phase cycle.   The accepted data are again those of Mr. Dugald Clerk, from experiment, as follows:

No.	Volume Gas.	Volumes Air.	Maximum Gauge Pressure, lbs. per sq. in.	Time of Explosion, Seconds.	Absolute Fahr. Temperature of Explosion from Observed Pressure.
2	I	13	52	0.28	1934°
4	I	11	63	0.18	2309
5	I	9	69	0.13	2525
6	I	7	89	0.07	3236
7	I	5	96	0.05	3484

Mixtures more dilute than 14 or 15 of air to one of gas fail to ignite. Six volumes of air furnish just about the required oxygen for complete combustion. The time of explosion can be shortened by adapting the shape of the vessel to more rapid propagation of the flame, or by igniting by the injection of a flame into the combustible mixture.

When the capacity of the mixture for resisting the cooling effect of the walls is considered, or its ability to keep up its pressure until a part of the stroke is completed, the best mixture is found between $\frac{1}{12}$ or $\frac{1}{14}$, or about one of gas to thirteen of air.

294. Thermodynamic Efficiency of the Otto Engine considered as a Carnot Engine.—If it be assumed that the heat caused by the explosive ignition of the gas in air is all imparted to the air at that temperature of explosion (T), and that by the compression either in pump or cylinder the temperature has been brought up to (T_c), the heat supplied to the cycle will be supplied at the constant volume v_c prevailing at the end of such compression, and will be

$$H_1 = C_v(T - T_c).$$

If the condition be assumed which is the most usual, that the final volume after expansion is the same as that at atmospheric pressure before compression was begun, then the heat necessary to bring the mixture back to its original state may be abstracted also at constant volume; hence if T_2 be the final temperature at exhaust, and T_a the atmospheric temperature before compression, the heat discharged will be

$$H_2 = C_v(T_2 - T_a).$$

Hence the Carnot efficiency will be

$$\frac{H_1 - H_2}{H_1} = \frac{C_v(T - T_c) - C_v(T_2 - T_a)}{C_v(T - T_c)}$$

$$= 1 - \frac{T_2 - T_a}{T - T_c}.$$

The two curves of temperature range being adiabatic, and the volume range being the same,

$$\frac{T_2}{T} = \frac{T_a}{T_c}.$$

Hence

$$E = 1 - \frac{T_2}{T} = 1 - \frac{T_a}{T_c}$$

$$= 1 - \left(\frac{v_c}{v_0}\right)^{n-1},$$

when v_0 is the final volume at the end of the stroke. This says, in other words, that the efficiency is greater as the ratio between the initial and final volumes of the compressing
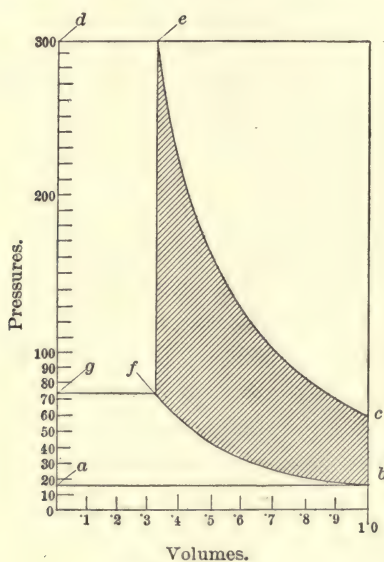
FIG. 163.

stroke grows less, or the greater the compression. The exponent n is the ratio between specific heats of air. Fig. 163 shows the perfect diagram corresponding to this design.

The fact that the theoretical efficiency can be increased by increasing the degree of compression, which would not be true in practice to the same degree, shows the danger of applying Carnot's reasoning to cases to which it is not properly applicable.

295. Temperature-entropy Diagram for the Gas-engine —The discussion of Joule's air-engine in § 274, where the heat medium receives its heat and increase of entropy at an increasing temperature, has opened the way to the discussion
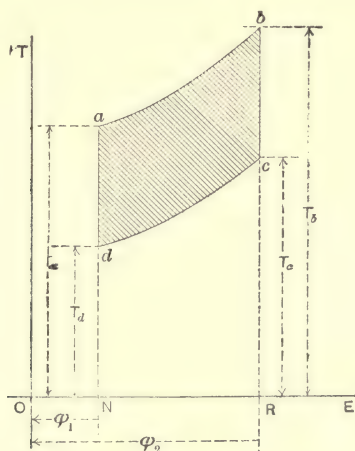
Fig. 164.

of the temperature-entropy diagram for the gas-engine. In its theoretical form, when operated with compression either in a separate cylinder or in the working cylinder, the adiabatic compression of the charge raises the temperature from d to a without increase of entropy. The ignition of the combustible gas at increasing temperature causes the entropy to rise according to the law

$$\phi = C_v \text{ hyp. log } \frac{T}{T_a}.$$

The expansion during the working stroke supposed to be adiabatic would drop the temperature down to that represented by the point c on the temperature-ordinate, at which the exhaust may be conceived to open. Then the curve cd is a drop in entropy and temperature at the constant pressure of the exhausting products of combustion, to close the cycle and bring back the condition of temperature and entropy belonging to the mixture as drawn into the compressing cylinder.

The difference between the gas-engine cycle and the theoretical Carnot cycle is made clear by observing that for the ideal Carnot cycle the line ad should have been carried up to meet a horizontal line through b which represents the maximum temperature attained; the entropy should increase at constant temperature. Secondly, the adiabatic expansion should lower the temperature till it descends along bc to meet a horizontal through the point d; the rejection must take place at the constant lower temperature. The relation of the shaded area to the total $T\phi$ area might thus be called the *possible* efficiency, while the ratio between the rectangle given by the lines through b and d as compared with the total area whose base is NR and whose height is bR might be called the ideal efficiency.

In drawing a temperature-entropy diagram for an actual engine from its indicator-diagram on the pv ordinates, the relation is used (§ 124) that the work done with isothermal expansion in foot-pounds should be equal to 778 times the heat-units supplied. With permanent gases this work will be expressed by

$$RT \text{ hyp. } \log r = RT \text{ hyp. } \log \frac{v_2}{v_1},$$

when the 778 is divided out to reduce it to heat-units. But the work will also be

$$T(\phi_2 - \phi_1),$$

whence the entropy change equivalent to the actual change
will become

$$\phi = R \text{ hyp. log } \frac{v_2}{v_1}$$

by dividing both members by T.

This can be written

$$\phi = (C_p - C_v) \text{ hyp. log } \frac{T_2}{T_1}$$

$$= C_p \text{ hyp. log } \frac{T_2}{T_1} - C_v \text{ hyp. log } \frac{T_2}{T_1}.$$

It follows, therefore, that points on the indicator-diagram
may be transferred to the temperature-entropy diagram by
the graphical expedient of drawing a line of constant pressure
through one point of a pv curve, and a line of constant
volume through another (Fig. 165). These two construction
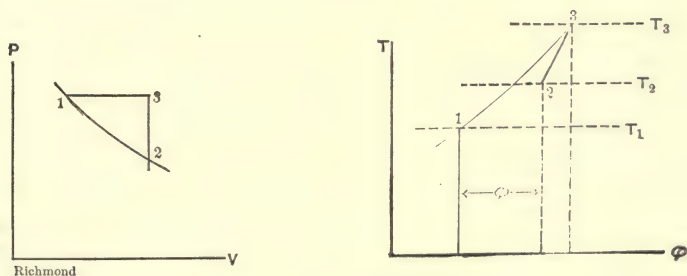
Fɪɢ. 165.

lines will intersect at a pv point which will have a corre-
sponding temperature T, established by the constant relation
$PV = RT$. Then if in the left-hand part of Fig. 165 the
points 1, 2, and 3 be the selected and resulting points, the
lines T_1, T_2, and T_3 will indicate the corresponding isothermal

lines.   Then the constant pressure curve 1- 3 will have a re-
lation between co-ordinates

$$C_p \text{ hyp. log } \frac{T_3}{T_1} = C_p \text{ hyp. log } \frac{v_2}{v_1},$$

since, the pressure being constant, the temperature will vary
as the volume.   The constant volume curve 3–2 will have a
relation between co-ordinates given by the expression

$$- C_v \text{ hyp. log } \frac{T_1}{T_3}.$$

By the expedient of multiplying both terms of the fraction
by $- \dfrac{T_2^2}{T_3^2}$, this becomes

$$+ C_v \text{ hyp. log } \frac{T_2}{T_3},$$

which (because the volume is constant, which again makes
the pressure proportionate to the temperature) may be written

$$C_v \text{ hyp. log } \frac{p_2}{p_1},$$

since the line through the point 1 is a constant pressure line
and hence $p_3 = p_1$.   Hence the equation for the entropy
becomes transformed from

$$\phi = C_p \text{ hyp. log } \frac{T_2}{T_1} - C_v \text{ hyp. log } \frac{T_2}{T_1}$$

into the expression

$$\phi = C_p \text{ hyp. log } \frac{v_2}{v_1} + C_v \text{ hyp. log } \frac{p_2}{p_1},$$

which can be used to transfer points on one diagram to the
other, provided proper values are found for the quantities
C_p and C_v for the mixture of air and gas during compression,

and for the products of combustion after ignition and during expansion. Mr. Geo. Richmond by a process of approximation has proposed values as follows:

$$C_p = \begin{cases} .246 \text{ on the compression curve} \\ .26 \quad \text{`` `` expansion `` } \end{cases}$$

$$C_v = \begin{cases} .176 \text{ `` `` compression `` } \\ .189 \text{ `` `` expansion `` } \end{cases}$$

Following this method, the pv diagram at the left of Fig. 166 gives the straight line 1–5 on the T.E. curve to correspond to the practically adiabatic compression line 1–5
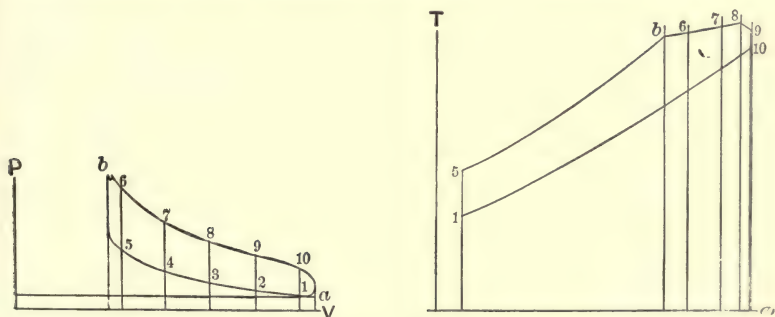
FIG. 166.

on the pv diagram. From 5 to b is the increase of temperature and entropy at constant volume, corresponding to the ignition rise in pressure from 5 to b on the indicator-diagram resulting from the ignition. Points 6, 7, 8, 9, and 10 on the indicator-diagram are found by the equation above; and from 10 back to 1, to close the diagram, a constant pressure curve corresponds to the drop in entropy and temperature incident to the exhaust, to reach the condition when compression is to begin anew. The calculation made by Mr. Richmond is given in the table on page 438.*

* School of Mines Quarterly, Columbia University, vol. XVIII. Jan. 1897, p. 146.

Point.	PV	$\dfrac{T_2}{T_1}$	$.176 \log P + .246 \log V$	$\phi_1 - \phi_2$
1	475	1.000	6094	. . . .
2	545	1.147	6149	55
3	600	1.263	6159	10
4	661	1.391	6154	— 5
5	706	1.486	6095	— 59
			$.189 \log P + .260 \log V$	
b	1391	2.928	6943	854
6	1404	2.957	7045	102
7	1438	3.027	7183	138
8	1460	3.073	7268	85
9	1428	3.006	7313	45
10	1300	2.800	7308	5

The table makes it appear that the changes in entropy from 1 to 5 or during the compression are so small that the curve is practically isentropic (§ 124) or adiabatic, for the scale used at any rate. This is doubtless to be explained by the compensatory action of cylinder-walls which are warmer than the mixture at the beginning of the stroke and cooler at the end. On the expansion curve, while the great rise in entropy corresponds to the ignition stage at b, yet the positive value (while a diminishing one) for the succeeding points means doubtless that the gas continues to burn with a delayed combustion till near the end of the stroke. The action of the cooling water-jacket no doubt masks also to some extent the real value of the heat added by this "after-burning" process.

The temperature-entropy diagram can also be made to show the gain in thermal efficiency resulting from the extension of the expansion down to atmospheric pressure. If the supposed theoretical diagram be again referred to, and the expansion be carried down to the lower temperature 5 (Fig. 167), instead of having the exhaust open at 3 before the pressure falls to atmosphere, then the point 5 is to be joined to 4

by a curve of equal pressure. This will be a lower or more gently sloping line than the curve 4–3, and the triangular area 5–4–3 will represent the thermal gain over the condition of incomplete expansion. It must not be overlooked, however, here as elsewhere, that this attempt to secure complete expansion is followed by wide variations in the propelling energy on the piston, and the bad effect of the irregularity
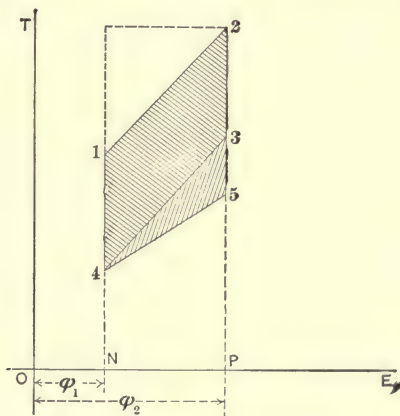
FIG. 167.

of motion may be of greater consequence than the thermal gain.

Furthermore, the temperature-entropy diagram may be used to make clear the effect of the water-jacket in the gas-engine. If the action of the water-jacket removes both temperature and entropy during the expansion, so that when the exhaust opens the state of the mixture is represented by a point located at 6 instead of at 3, then the triangle 2–3–6 (Fig. 168) shows the quantity of heat so disposed of. The effect of the jacket thus appears to be to diminish the heat swept away by the exhaust by the quantity represented by the area under the line 3–6, which would be exhausted with the products of combustion if there were no jacket. The sup-

pression of the jacket-loss would not result in its conversion into work. The jacket would act to bring the expansion line lower than an adiabatic on the *pv* plane. If, however, the phenomenon of retarded combustion be in action, the expansion pressure would be increased; and while perhaps a less temperature would be attained, a greater value for the entropy change might be made to result. That is, if, instead of having the maximum temperature represented by 2 reached by the ignition (Fig. 169), a temperature of 7 only
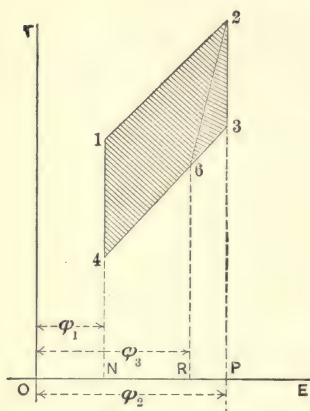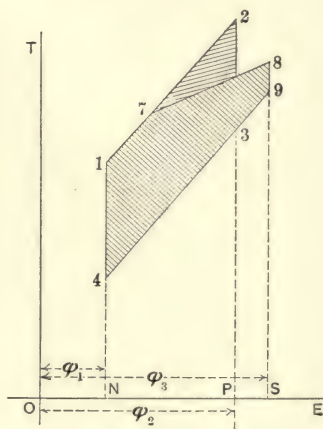
Fɪɢ. 168.   Fɪɢ. 169.

is reached by the ignition at the beginning of the working stroke, then the retarded combustion traces a path such as 1–7–8 during the expansion whose latter part only is adiabatic. If the area of heat represented by the area whose upper boundary is 1–7–8 is the same as that bounded by the line 1–7–2, then the combustion is completed before the exhaust opens and no waste of gas occurs. If less, then the retarded combustion has been so noticeable as to cause a waste of unburned gas.

296. Compound Gas-engines.—It will have been observable from most of the indicator-diagrams of gas-engines that

the exhaust opens while a considerable pressure remains upon the piston. This may be as much as 30 to 40 pounds above the atmosphere at the beginning, and the coughing noise of such exhaust is a great objection to this type of engine for many locations, apart from the apparent waste of available energy.

The problem may be stated generally, by saying that it is desired to expand the compressed charge during the working stroke to a volume greater than that which the charge occupied before compression. This cannot be done in the single cylinder of the Otto cycle-engine, and the attempts to secure continuous expansion in more than one cylinder have not been successful as yet. In some two-cylinder engines one has had a larger volume than the other, either by different areas or different length of stroke, but the two cylinders have received their charge simultaneously and not in succession as a continuous process. Clerk has suggested that in small engines it is to be questioned whether an increase in the expansion beyond the volume of the charge before compression may not so far reduce the volume of mixture dealt with at each stroke as to increase the relative loss of heat to the jacketed cylinder-walls to a point where the gain from greater expansion will be more than neutralized. In larger engines dealing with larger volumes of gas and with their slower transfer of heat to metal walls, the gain from successful compounding will be realized when further investigation shall have revealed some principles as yet not mastered.

297. The Oil-engine, using Kerosene or Non-volatile Oils.—It will be at once apparent that if the object in the gas-engine is to burn the fuel supplying heat directly in the working cylinder, this same object can also be secured by injecting a liquid fuel in a finely divided state, or a state of vesicular vapor, mixed with the air which it requires for its combustion. Hence it was early proposed and tried to use

petroleum and its derivative oils, so as to make this type of motor available where municipal or private gas companies were not at hand, or where the manufacture of gas by producer methods was not convenient or practicable. The first practical petroleum engine was that of Julius Hock of Vienna (1870), and the early Brayton engines of 1873 were planned for oil. If the more volatile derivatives of petroleum are used, such as naphtha or gasoline, a current of air passing over or through them will absorb hydrocarbons to such an extent as to become an illuminating-gas, and can be used as such in an engine (§ 47). But, as a rule, the dangers connected with the use of these lighter oils, and particularly with their storage in any quantity, have precluded their use on any large scale. On the other hand, the use of crude petroleum is attended with difficulty from the presence of its heavier and less inflammable constituents, whereby it becomes difficult to burn it completely and to ignite it promptly. There are troublesome residues also, which leave a deposit in the cylinder (§ 49). In many places, furthermore, the use of crude oil is prohibited by ordinance by reason of its offensive odor, and from the danger connected with the presence in it of its volatile elements. Hence the usual oil-engine burns oil of the grade known as burning oil, or kerosene (§ 43), which is safe and efficient.

The use of a liquid fuel suggests that it should be first finely subdivided or made into a spray or mist by the action of a current of compressed air, and then afterwards the true vaporization of the liquid oil shall take place by heat. In certain successful oil-engines, however, the spraying process is omitted, and the oil injected in its liquid state. Oil-engines may therefore be classified into groups:

1. The oil is atomized or sprayed, and then vaporized.
2. The oil enters the working cylinder as a liquid, and is there vaporized.

3. The oil is vaporized in a separate vessel, from which it passes to the working cylinder as a gas.

As in the case of gas-engines, the methods for igniting the mixture of air and oil-vapor will distinguish oil-engines from each other.

The mixture may be fired by (1) an electric spark; (2) by passing through an incandescent tube; (3) by heat of the internal surfaces, combined with the heat of compression.

The Priestman engine will serve as a type of the first group; the Hornsby-Akroyd of the second; and of the third group the best known examples are the British designs of Crossley and of Fielding and Platt.

In the Priestman engine the air-supply is passed through a heating-chamber, which is surrounded by the hot exhaust-gases. If the latter are at 600° F., the chamber will be at nearly 300° F. The oil is injected into this chamber by a small jet of air, and changes from spray to oil-vapor by evaporation, and passes into the cylinder upon the aspirating stroke. The compression of the preheated air brings it to a high temperature, above that to be met in the gas-engine, thus lowering the weight of charge present in the engine at each stroke, and reducing the average available pressure. The high temperature renders the mixture liable also to premature ignitions. The presence of the charge in the vaporizing chamber in the event of a back explosion from the working cylinder constitutes a menace to the safety of the whole machine. The necessity for keeping the vaporizing chamber hot by means of exhaust-gases makes it necessary to use some oil at every stroke, even at light loads, so that governing forms a difficulty with the design, and the consumption of oil is not proportionately diminished as the load diminishes.

In the Hornsby-Akroyd engine the vaporizer is a cast-iron non-jacketed chamber behind the working cylinder, to which it is connected by a narrow neck. The heat due to each ignition keeps this chamber hot enough so that oil

injected into it is vaporized, and the completion of the compression stroke, forcing in a charge of fresh air from without, raises the mixture to the point of ignition, and it ignites without outside means to effect it. The vaporizer is heated by gas or by a blast oil-lamp outside of it to start the engine. Governing is effected by causing the oil-pump to send its charge back into the suction by a by-pass, instead of into the vaporizer, but, as in the previous case, the vaporizer must not be allowed to cool off too much at light loads, else ignition fails.

This type suffers also from the lower average pressure than is secured with gas, because the mixture of oil, gas, and air is less likely to be thorough, and an excess of air is therefore usual to secure perfect combustion. The engine must furthermore be massive to meet the possibility and likelihood of double charges, when for any cause an ignition has been missed on one stroke, and twice the normal quantity of oil is fired at the next ignition.

In the third group the typical engine has the oil and a small volume of air injected into a heated vaporizer consisting of tubes or passages kept hot by waste heat from the blast-lamp which heats the ignition-tube. A second volume of air, heated before it enters, joins the oil in the vaporizer and completes the vaporization. The main charge of air for the mixture is not preheated, but enters through a separate valve, mixing with the oil-vapor and being ignited by an incandescent tube. The use of cool air permits higher compression, and hence higher mean pressures, with the attendant advantages. The exterior lamp with its heat and noise are objections, however, and the simplicity of the Hornsby-Akroyd design has much to commend it.

298. The Oil-engine using Gasoline or Light Volatile Oils.—The difficulties from legal restrictions and city ordinances attaching to the storage and use of the volatile oils have been already referred to in a preceding paragraph.

Their lightness in weight, and the lightness of the fuel-supply in tanks for a given traverse, have made this type popular on the continent of Europe for motor-carriages or automobile use. By reason of the volatile character of the liquid fuel, it is only necessary to force air through and over it, for the former to take up hydrocarbon vapor in quantities to make an ignitible gas. The mixture is easily ignited by passing it through incandescent tubes heated by a shunt oil-circuit which supplies a lamp beneath them, burning with a Bunsen flame. Electric igniters are not used for motor-carriages by reason of the weight of the necessary battery.

The best known gasoline-motor in Europe is the Daimler. The usual form has two cylinders, closed at both ends. The back ends behind the pistons operate on the usual Otto cycle, the front ends acting as air-pumps, so that an additional pressure of a few pounds is secured before ignition with a surcharge of air from the front ends. The compressed air from the front effects the carbonization of the working charge, and also operates the lamp of the ignition-tubes. These are two platinum tubes. The two cylinders incline to each other at an angle of about 30°, with their connecting-rods acting on a common crank-pin. In motor-carriages the main shaft runs at a constant speed and in one direction, so that speed changes and reversing is effected by gears and clutches.

The naphtha-engine, as used extensively in launches, is not a direct-combustion engine, but uses the volatile liquid in a closed vapor cycle as steam is used in the condensing engine. This will be referred to under "Vapor-engines."

299. The Diesel Petroleum-motor.—A most interesting development of the principles of the heat-engine has been made in Germany during the years 1893–97 by a Mr. Rudolf Diesel. The fundamental peculiarity of his engine lies in the same lines as those of the Hornsby-Akroyd. The idea is to compress the air to a point at which its temperature due to the compression (§ 182) shall be above the point of ignition of the fuel to be used. If the latter is petroleum, crude or

refined, a measured quantity of it as controlled by a governor is injected into the highly heated air. It enters as a vapor by being sprayed by air from a small auxiliary pump, and by being forced also through a close-mesh wire gauze. If the compression has been carried (as is expected) to a pressure of 500 pounds per square inch or over, the finely divided oil ignites, and supplies to the expanding air the heat equivalent to the work it is doing in expanding, realizing (to the extent that it secures this result) the ideal Carnot theorem of having the heat supplied isothermally during expansion or at the constant highest temperature of the cycle instead of having the supply occur along a line of falling temperature. It would be obviously possible to supply fuel enough to do all the work of the stroke without drop of temperature, but this would consume more oil in the first place, and in the second place would leave the gases when ready to exhaust at a temperature unnecessarily and wastefully high. Hence the charge of oil is proportioned so as to keep up the expanding gases to T_1 during a part only of the stroke, which results in there being always an excess of air so as to secure complete combustion. When the oil-charge is completely burned at an early period of the working stroke and can furnish no more heat to the expanding air, cooling begins by the adiabatic expansion, which then sets in, and which continues till the end of the stroke, leaving the gases to exhaust at a lower temperature, and without causing so much heat to pass out by way of the jackets. The latter are retained, however, for the sake of uniformity of temperature in valves and pistons. A new charge is again drawn in on an aspirating stroke, and compressed on the return stroke as in the Otto cycle.

The injection of the fuel *after* compression is complete eliminates the danger of premature ignition and explosion; governing is also much simplified; and what is more important than all, a practically perfect combustion is secured, without carbon loss in smoke or in lamp-black deposit.

Starting is effected by a few strokes by a hand oil-pump,

to throw a preliminary fuel-supply into the casing of the fuel-valve; from a storage of compressed air at high tension from a previous run a charge of cold air is admitted by a starting-valve to the working cylinder, sufficient to turn the fly-wheel through two revolutions, or past one compressing stroke. This is done with the valve-cams out of gear, but at the close of the first revolution the cams are thrown into gear automatically, and the engine will then start off. The high temperatures used enable this engine to work on low-grade oils, or even upon powdered solid fuel, as well as with kerosene, alcohol, or gas. Governing is done by by-passing the charge or injection from the oil-pump, which is proportioned for the maximum supply to be required.

By applying the temperature-entropy diagram to the cycle of the Diesel motor, it will be apparent that when the proper practical conditions are secured for it, it approaches more nearly to the ideal Carnot conception than any of the preceding motors, and therefore the ratio of heat utilized to heat supplied or its efficiency should be by so much greater as this approach is more close. Tests show an efficiency of 38 per cent for the Diesel engine, as compared with 25 to 30 in ordinary gas engines. Fig. 170 illustrates the engine in section, and Fig. 171 is a reproduction of one of its cards.

300. **Performance and Economy of Direct-combustion Engines.**—Any comparison or critical discussion of different types of direct-combustion engine when stated in consumption of fuel per H.P. must be unsatisfactory and unreliable to the extent that the heat-units per unit of fuel, or from the analysis of the fuel, are unknown or unstated. The figures following are therefore given only as guides, and as points in a reconnaissance survey, by reason of the lack of such determining data, and because also the sizes of the engine are not given. Large engines burn less gas per H.P. than small ones.

With the ordinary city illuminating-gas to be met in Great Britain and the United States an average ranging from 20 to
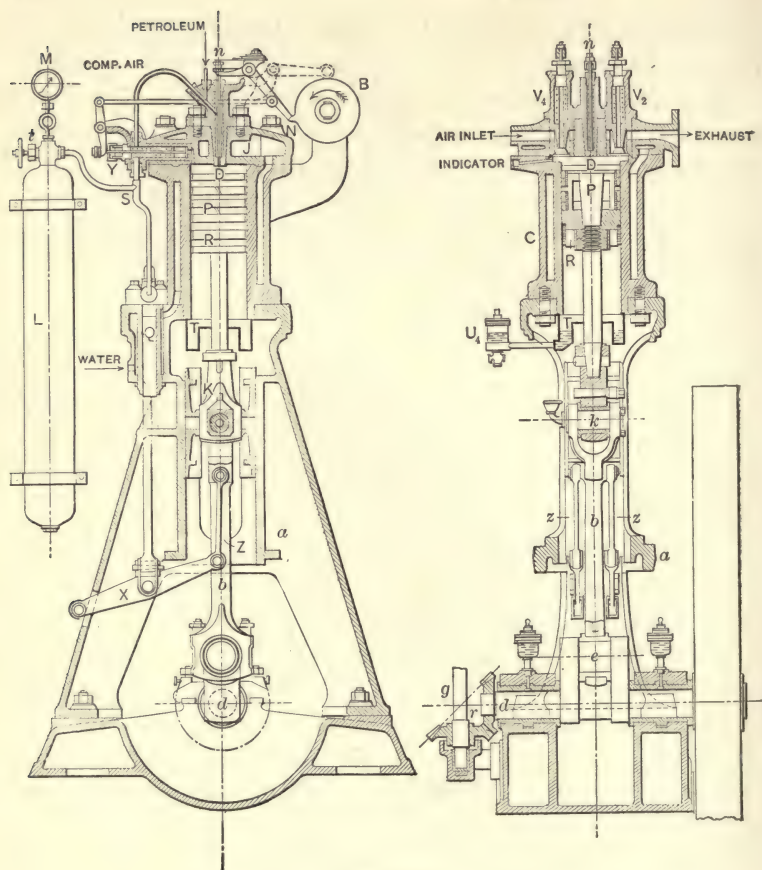
FIG. 170.
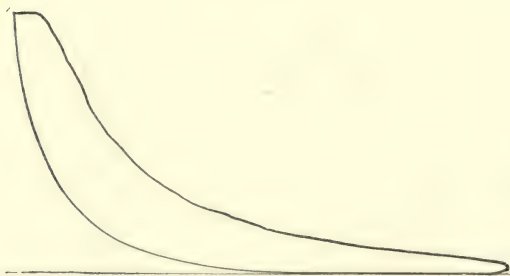
FIG. 171.

24 cubic feet per indicated horse-power may be expected, which corresponds to a net, or brake, horse-power consumption of 24 to 30 cubic feet per hour. Record consumptions run as low as $13\frac{1}{2}$ to 15 cubic feet per I.H.P. per hour, or 18 to 20 cubic feet per brake horse-power.

The largest gas-engines of Europe and probably in the world are of 400 H.P.; the highest number of revolutions per minute is 400.

When gas can be secured from producers of the Dowson or Taylor or other economical type the lower calorific power per cubic foot increases the consumption in cubic feet, but not the consumption per pound of coal or of carbon. For example, in a town in Connecticut where electric lighting is supplied by gas-engines operating on producer-gas the gas has only 153 B.T.U. per cubic foot, and 99 cubic feet are required by brake horse-power per hour. This corresponds, however, to a consumption of 1.075 pounds of carbon per brake horse-power per hour. English and Continental records are 1.31 to 1.34 pounds of coal per I.H.P. per hour on small engines, and an opinion has been expressed that with engines of over 100 H.P. the consumption should drop to one pound.

With oil of the grade of kerosene the consumption in pounds per I.H.P. per hour ranges from 0.75 to 1.25, corresponding to a consumption per brake horse-power ranging from 0.82 to 1.68 pounds. The Diesel motor has an authenticated record of 240 grams of kerosene oil per brake horse-power per hour, equivalent to 0.531 pound, which, when reduced on the basis of a price of $2\frac{1}{2}$ to 3 cents per gallon for fuel-oil, results in a calculated consumption of 15 cents' worth of such oil per horse-power per hour in engines of 100 H.P.

301. Advantages of the Gas- or Oil-engine.—The principle of direct combustion, or the liberation of the heat energy directly in the working cylinder, as contrasted with the indirect method of imparting this heat energy to the heat

medium in a separate vessel, offers some inherent advantages; and certain others attach to the methods used to avail of this principle or are incidental to it. That is, there are some advantages attaching intrinsically to the use of gas or oil as a source of heat energy (§§ 42–56) apart from the principle of direct combustion as a means of utilizing that energy.

Attaching to the direct-combustion principle may be noted:

(1) The direct utilization of heat energy to drive the piston in a single apparatus.

(2) Resulting from this an economy of combustible per horse-power per hour because heat is not wasted in a furnace or chimney, or in doing work upon a heat medium which is not utilized in the engine.

(3) No fuel is consumed wastefully to start up the motor, nor is any wasted after the engine stops. The losses in banking fires under a boiler which runs intermittently are avoided, and the losses of fuel in the grates from cleaning and when the run is over.

(4) The engine is ready to start on the instant, without delay caused by getting up steam or starting the fire.

(5) By direct use of gas or oil in engines the advantages of storage of energy are reaped. Gas can be made when convenient, and stored in holders for use after working-hours or when the generating-plant is not running.

(6) Incident to this is the advantage of subdividing power units in a large plant, each of which may receive its supply of motor energy through pipes without loss, and which can be run independently of each other as to time, speed, capacity, and the like, as long as the store of gas or oil holds out.

(7) Storage of energy in the form of gas under pressure enables great power to be stored in small bulk and with small weight, to be expended in motors as required. The automobile and the experimental flying-machine avail of this

feature. So also might street-cars where electricity or compressed air is not available or preferable.

(8) The absence of the boiler and its furnace and chimney make the plant compact when the gas generation and holders are provided for elsewhere. The oil-engine is portable from the same causes, and convenient for temporary erection.

(9) The absence of the boiler (when not offset by the presence of the producer or the gas-holder) lowers the insurance risk, and the owner avoids the expense of licensed attendants upon the steam-generating plant. At least this is the case where local legislation is in force on these points, as in most cities.

(10) The absence of the boiler, with its furnace and chimney, avoids the repair and maintenance expenses which attach to these features of a power plant, as well as the labor to operate them, and their first cost.

The gas- or oil-engine furthermore attaches to itself the advantages of mechanical stoking and of oil or gas used as fuel. Such are:

(11) Combustion practically perfect and smokeless.

(12) No human labor for handling fuel into the heat apparatus nor for disposing of the ashes, with attendant cost.

(13) No dust nor sparks nor soot, except at the central gas-generator plant. With oil there is none anywhere.

302. Disadvantages of the Gas- or Oil-engine.—As in the foregoing paragraph, some of the following objections are inherent, and others attach only to certain solutions of the problem. An engine is excellent to the extent that it avoids these difficulties.

(1) Gas-engines are usually single-acting, if not acting on only one stroke in four. This makes them more bulky than the double-acting steam-engine using the same mean pressure.

(2) It is difficult to command a high mean pressure.

(3) The effort is irregular, and hence a disproportionately heavy fly-wheel is demanded to secure steadiness.

(4) Where governing is done by missing a charge, the speed must vary widely. Many types do not govern well nor closely.

(5) Most require an auxiliary of some sort to start them from rest.

(6) There is no way of increasing the power of the engine beyond the maximum for which it is proportioned, to meet short demands for greater power.

(7) It is not usually arranged to run in both directions, or is not easily reversible.

(8) An unpleasant odor of partly burned or of unconsumed gas belongs to many examples. Leakage seems unavoidable.

(9) The high tension of the contents of the cylinder at the end of expansion makes the exhaust noisy, or like a succession of coughs, and there is a loss of heat in such hot gases.

(10) The water-jacket around the cylinder is necessary to keep it and the valves cool enough to be tight and prevent deformation by heat and wasting from oxidation. The heat which goes into this water-jacket would otherwise go into the exhaust (§ 295), and could not be saved except at expense in other ways; but the engine must be supplied with the water for this jacket, and with provisions for the water to cool if it is too valuable to be wasted.

(11) If the water-jacket is ineffective, and often where it works well, the lubrication of the hot-piston is difficult. Packings should be avoided, or should be metallic when they are required.

(12) Explosions of some violence occur in the exhaust-pipe. The fire laws of New York compel the metallic exhaust-pipe to be carried to free air, and not merely to enter a brick flue leading to the air. Explosions in such flues would be disastrous. The explosion is due to a charge which is not ignited in the cylinder.

(13) Lamp-black deposits, the result of imperfect combustion, clog and defile the working parts, valves, ports, and

the like.   This deposit is very troublesome between the
points of spark-igniters.   The difficulty can only be avoided
by having the contacts self-cleansing by sliding upon each
other.

   303. Conclusion.—The gas-engine has been much more
studied in England and the Continent than in the United
States.   In most places it is cheaper to burn city gas in gas-
engines to operate dynamos to generate electric current for
incandescent lighting than it is to burn the gas directly in
chandeliers.   Where fuel-gas can be made in producers this
plan is much the cheaper.   In many cities the insurance rate
upon the producer plant makes its cost prohibitory.   The
gas-engine (or oil-engine) is the rival which the steam-engine
is to anticipate in the next few years, for small sizes in any
case, and possibly for units of considerable size also.

   Fig. 173 shows the successive positions of the usual form
of slide-valve in Otto gas-engines, and should be studied in
connection with Fig. 144 on page 414.
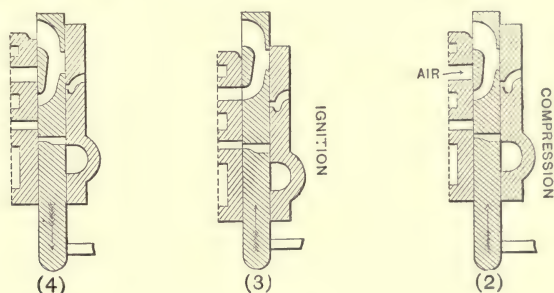
FIG. 173.

CHAPTER XXI.

VAPOR-ENGINES.

305. Introductory.—It has been already noted (Chapter IX) that any other convenient heat media could be used to carry the heat energy from the furnace or source of heat and make it available for doing of mechanical work in the engine-cylinder, if there were reasons why steam or air should not be preferred. It was further suggested, however, that there were practical reasons why the effort to use other media had not as yet been approved by commercial success. The object of this chapter is to go more fully into this question with respect to a selected series of heat-carriers, and to show the physical and theoretical basis for the disapproval of their claims for recognition for these uses, except perhaps in a few special cases.

Water and air being the only heat media found in a natural state in unlimited quantities, all the others are manufactured products or compounds which must be purchased. Hence they must be worked in a closed cycle, and not rejected to the atmosphere from which they cannot be regained. They will therefore be operated in condensing engines, and will be used as vapors to be condensed from the gaseous to the liquid state after working in their cylinder. The heat will be applied to convert them into gases, becoming latent in the process of increasing their entropy. This heat will be given out in adiabatic expansion in part, and in part to the condensing appliance, as discussed in §§ 195 and 218 *et seq.* of Chapters XIV and XV, and the cycle will be

closed by pumping the liquid back into the generator to be raised in temperature and entropy to the high initial condition again.

The attraction to the more volatile vapors of certain liquids comes about from the facts of their physical constitution, whereby, either by reason of a lower specific heat or for molecular reasons, they are raised to a higher tension, as registered by a pressure-gauge, by a smaller amount of heat than is the case when heat is applied to water to make it into steam. If gauge pressure meant heat energy also, then there would be no question as to the thermal superiority of these more volatile media. It will be the object of the succeeding paragraphs to show how insignificant or negative these thermal advantages usually are.

The problem of the use of volatile vapors may be approached from several different points of view. It may be first examined from the assumption that the competing vapor is to be worked within the same temperature limits as the accepted steam-engine, disregarding the inconvenient pressures which result, and the relation of work done to heat supplied can be deduced under these conditions.

Or, secondly, the problem of keeping the pressures within normal limits may be made the determining one, and the temperatures adjusted to the condition of strength of cylinders and other parts which must resist pressure strain.

Or, thirdly, a set of conditions advantageous to the use of the vapors may be assumed, and the computations directed to show whether the resulting quantity of the heat medium by weight or the size of the cylinder by volume is more advantageous or economical of fuel than in the case of the steam-cylinder.

306. Formulæ for the Work of a Vapor.—The accepted formulæ for the work of a vapor which expands isothermally with change of entropy at the constant temperature T_1, as represented by the line AB in Fig. 175; then expands adia-

batically along the curve BC to a final pressure CG, or p_2; drops to a back-pressure GH, or p_3, and is then worked back by isothermal compression and finally by adiabatic compression to the origin A, has been deduced by Rankine, Clausius, and others from differential equations, and appears in three
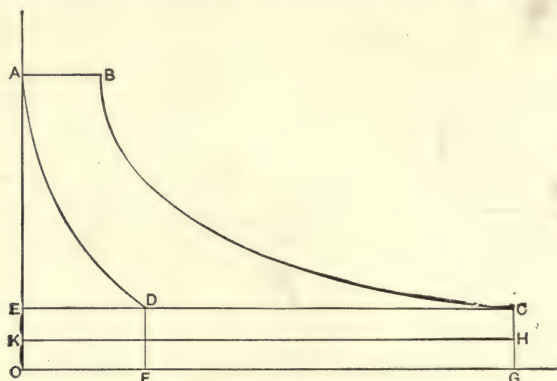
FIG. 175.

forms. If the cubic foot is used as the unit, then the work of one stroke in foot-pounds will be:

$$\text{Work} = JC_pD_1\left\{ T_1 - T_2\left(1 + \text{hyp. } \log \frac{T_1}{T_2}\right) \right\}$$
$$+ L_1\left(\frac{T_1 - T_2}{T_1}\right) + r(p_2 - p_3),$$

in which $J = 778$, the mechanical equivalent of one B.T.U.;

$\quad\quad C =$ the specific heat, at constant pressure, taken
$\quad\quad\quad\quad$ at its mean value between T_1 and T_2, if it is
$\quad\quad\quad\quad$ variable;

$\quad\quad D_1 =$ weight in pounds of one cubic foot of the vapor
$\quad\quad\quad\quad$ at the temperature T_1;

T_1 and $T_2 =$ absolute temperatures corresponding to the
$\quad\quad\quad\quad$ pressure values used;

$\quad\quad L_1 =$ the latent heat of evaporation of a cubic foot
$\quad\quad\quad\quad$ of the vapor at the temperature T_1;

p_2 and p_3 = the pressures at the end of adiabatic expansion and in the condenser, respectively;

r = the ratio of expansion to reduce the temperature of the vapor from T_1 to T_2. This is given by Rankine in the equation

$$r = \frac{AB}{EC} = \frac{T_2}{L_2}\left(778D_1 \text{ hyp. log } \frac{T_1}{T_2} + \frac{L_1}{T_1}\right),$$

in which L_2 = latent heat of evaporation of a cubic foot of vapor at the final temperature T_2.

With complete expansion, so that p_2 equals p_3, the last term disappears, or the area will be

$$\text{Work} = JCD_1\left\{ T_1 - T_2\left(1 + \text{hyp. log } \frac{T_1}{T_2}\right)\right\} + L_1\frac{T_1 - T_2}{T_1}.$$

If it be preferred to discuss the problem from the point of view of a *pound* of the vapor instead of a cubic foot, the expression for the work may be used in the more convenient of the two forms deduced by Rankine; that is, the formula may be, when p_2 is equal to p_3:

$$\text{Work} = ABCD = 778\left\{ T_1 - T_2\left(1 + \text{hyp. log } \frac{T_1}{T_2}\right)\right\} + \frac{T_1 - T_2}{T_1}L_1,$$

in which L_1 is the latent heat of evaporation of one pound of vapor at the temperature T_1 and is given by a formula

$$L_1 = a + bT_1,$$

or

$$L_1 = a + bT_1 + cT_1^2,$$

in which both b and c may have a negative value.

When there is a difference between the pressures p_2 and p_3 the formula may be transformed into

$$\text{Work} = ABCHK = a \text{ hyp. log} \frac{T_1}{T_2} - b(T_1 - T_2) + rv_1(p_2 - p_3),$$

in which v_1 is the volume occupied by one pound of the vapor at the pressure corresponding to T_1. Instead of rv_1, the quantity v_2 could be used, or the final expanded volume which is r times the initial volume admitted.

The next step in the problem is the demand upon the physicist or laboratory experimenter for the values of the factors or constants which enter the formulæ.

307. Experimental Data for a Problem in Vapors as Heat Media.—The data of density, pressure corresponding to temperature, latent heat at different temperatures, and specific heat for the vapors, have in most cases been reduced to formulæ with constants to be substituted, and these constants or factors are to be multiplied by the variable or arbitrary temperature. It will be necessary, therefore, to assume a temperature range between the limits of which the vapor is to be worked. The investigation, along these lines, whose completeness has made it almost a classic among American researches in this field is identified with the names of Messrs. Henry L. Gantt and D. H. Maury, and was first made in 1883–84. It was later revised by Prof. De Volson Wood. It will therefore be convenient to use the same assumptions and data which they have followed as respects temperature and the like, so as to follow a similar procedure.

If the standard steam-pressure range be taken as a starting-point to determine the temperature range, it may be taken as between 125 pounds per square inch and 10 pounds per square inch—absolute pressures counted from vacuum as a zero. This gives the temperature range

$$t_1 = 172° \text{ C.} = 342° \text{ Fahr.} = 802° \text{ abs. Fahr.};$$
$$t_2 = 90° \text{ C.} = 194° \text{ Fahr.} = 655° \text{ abs. Fahr.}$$

The Rankine formula for pressures with temperatures given is (§ 134)

$$\log p = A - \frac{B}{T} - \frac{C}{T^2},$$

in which A, B, and C are constants having the following values:

	A	$\log B$	$\log C$
Steam	8.28203	3.441474	5.583973
Alcohol	8.68170	3.4721707	5.4354446
Carbon disulphide....	7.4263	3.3274293	5.1344146
Chloroform	4.3807	[This B is 3.288394 negative]	6.1899631
Ammonia	8.4079	3.341632	....

Making the substitutions, the following values for the pressures result:

	p_1 Lbs. q. Ft.	p_2 Lbs. per Sq. Ft.
Steam	17408	1469
Alcohol	36450	3279
Carbon disulphide....	36745	7269
Chloroform	24871	5432
Ammonia	468700	113100
Ether	55350	1440

The specific heat at constant pressure for any temperature t for such vapors is given by a formula of the form (§ 139)

$$C_t = A + 2Bt + 3Ct^2.$$

The constants will be for the various vapors, if t is the temperature in degrees F.:

	b	c	d
Steam..................	0.99957333	0.000002222	0.0000000926
Alcohol.................	0.50954300	0.00056407	0.000000617284
Carbon disulphide	0.2323140	0.00004555	0.00000000000
Chloroform	0.2305470	0.00002817	0.00000000
Ammonia...............	....	....	....

Substituting the corresponding values, the mean value of C_p between T_1 and T_2 becomes

	Steam.	Alcohol.	Carb. Dis.	Chloroform.	Ammonia.
$C_p =$	1	0.954	0.257	0.245	1.229

The general formula for the weight of a cubic meter of a saturated vapor of the class under consideration (§ 141) is given by the formula

$$D_1 = \frac{1}{v_1} = \frac{p_1 \delta}{36.2 \sqrt{T_1}},$$

in which δ is the density of the vapor referred to that of air, and has been observed for the vapors in question to be:

	δ	D_1	D_2
Water..........................	0.6219	5.07	0.467
Alcohol.................	1.5819	25.18	2.74
Ether..........................	2.5573	78.66	14.47
Bisulphide of carbon	2.6258	45.37	9.98
Chloroform......................	4.1244	57.84	11.13

when the calculations are made for D_1 and D_2 by substituting in the formula. Or the following method may be used.

The latent heats of evaporation per kilogram (§ 140) at any temperature t is given by a formula of the form

$$r = a - bt - ct^2,$$

in which a, b, and c are constants having the following values (reduced to British units from the corresponding metric values determined by Regnault):

	a	b	c
Steam........................	1121.7	0.6946	0.00002222
Alcohol.......................	524.07	0.92211	− 0.000679
Carbon disulphide.............	164.57	0.0716	0.0002746
Chloroform	123.6	0.093	0.000282
Ammonia......................	555.5	0.613	0.000219

By substitution of the values of the constants a series of values for t_1 and t_2 result and which give the tabular values for L_1 and L_2 (the latent heat per pound) when these are multiplied by the weights per pound and by the symbol 778 or J to reduce heat-units to foot-pounds. Hence

	r_1 (Thermal Units.)	r_2 (Thermal Units.)	L_1 (Foot-pounds.)	L_2 (Foot-pounds.)
Steam...............	873.59	978.44	679653	761226
Alcohol.............	288.19	370.73	224216	288428
Carbon disulphide....	108.07	140.34	84077	109188
Chloroform..........	58.92	94.94	45843	73867
Ammonia	320.54	428.34	249383	333246

The weight of one cubic foot of saturated vapor is the ratio of the latent heats per pound (L) to the latent heats per cubic foot (l), or

$$w = \frac{l}{L}.$$

The latent heats per cubic foot in foot-pounds are:

	l_1	l_2
Steam...............	186905	20553
Alcohol	381359	43791
Carbon disulphide ...	259844	64970
Chloroform..........	136930	53358
Ammonia............	2954108	873561

Combining these with the values for L above,

	w_1 (Weight in 1 cu. ft.)	w_2 (Weight in 1 cu. ft.)
Steam...	0.275	0.027
Alcohol..............	1.701	0.151
Carbon disulphide ...	3.091	0.595
Chloroform	2.978	0.722
Ammonia............	11.842	2.621

308. Efficiency of a Volatile Vapor between given Temperature Limits. — The simplest case will be the hypothesis of Carnot, that all heat is imparted at the high temperature t_1 and withdrawn at the lower; the expansion and the compression adiabatic. To make the case general, suppose the heat L_1 to produce a volume v_1 which is not the unit volume whose weight is D_1. Then the work of expansion will become

$$W = JCD_1v_1 \left\{ T_1 - T_2 \left(1 + \text{hyp. log} \frac{T_1}{T_2} \right) \right\} + \frac{T_1 - T_2}{T_1} L_1 + v_2 p_2,$$

and the work of compression will be

$$W' = JCD_1v_1 \left\{ T_1 - T_2 \left(1 + \text{hyp. log} \frac{T_1}{T_2} \right) \right\} + p_2 v_2.$$

The effective work will therefore be what the previous discussions of Chapters XIV to XIX would have led to expect, or

$$(T_1 - T_2) \times \frac{L_1}{T_1} = (T_1 - T_2) \times (\phi_2 - \phi_1).$$

Substituting in these equations their values, the following tabular solution results:

Heat Medium.	Work of Expansion, Foot-pounds.	Work of Compression, Foot-pounds.	Difference Effective Work.	Heat used.	Efficiency, Per Cent.
Water	50323	15932	34391	186905	18.4
Alcohol..............	61043	26652	34391	186905	18.4
Ether	115000	80609	34391	186905	18.4
Bisulphide of carbon.	64538	30147	34391	186905	18.4
Chloroform	71843	37455	34391	186905	18.4
Ammonia	72222	37831	34391	186905	18.4

In discussing the above it will be noted:

(1) That the thermal efficiency of all the media when operated within the same limits of temperature is the same.

This might have been foreseen, since the formula indicates that the efficiency is independent of the medium used.

(2) The amount of heat required to evaporate a cubic
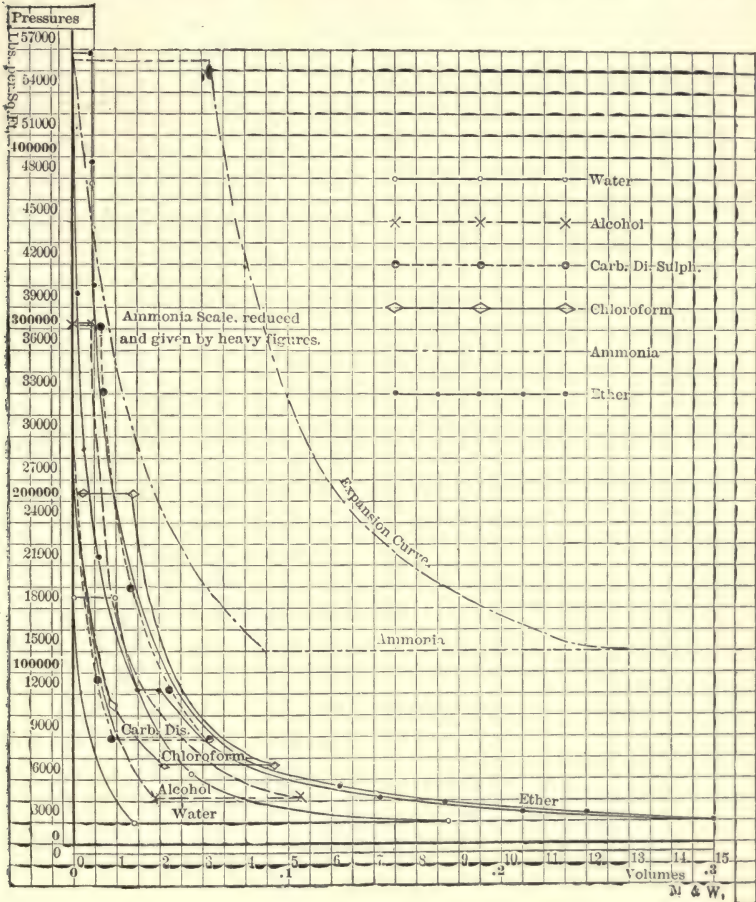
FIG. 176.

meter of water at T_1 degrees absolute is less than that required to evaporate the other vapors at that temperature. Note the values for L_1.

(3) Note also the values for L_2. The water in expanding

has given up more of its heat in doing work down to the temperature T_2 than has been given by the others. In other words, water has done the same amount of effective work as the others, and has left a less quantity of heat to be taken care of by the condensing water.

(4) The pressure of the vapor of water at the temperature T_1 is much less than that of the other vapors. Hence the others require much more massive castings and greater strength in all parts to withstand the strain than are needed for the steam-engine, and yet they do no more effective work in the same range. This pressure range for the same temperature range is shown most clearly in the plotted diagram (Fig. 176), which shows equal work areas, but with very different values for p_1 and v_2.

(5) The same diagram shows also the inconvenient losses from excessive back-pressures with the vapors other than water, which back-pressures, as above, can only be mitigated by a use of inconvenient volumes of condensing water at a low temperature, as will be referred to hereafter.

(6) The component works of expansion and compression differ very widely, while having a constant total difference. The discussion is therefore inconclusive in so far as any effect of varying interchange of heat with the cylinder-walls may be introduced by these differing states of the medium, in its cycle.

(7) The diversity in pressure range for the same temperature range turns the thoughts to the possibility of a species of compounding with two media in series. This will be referred to again.

309. Efficiency of a Volatile Vapor between given Pressure Limits.—The difficulties of the preceding propositions with respect to pressure make it desirable to examine the action of volatile vapors when the convenient pressures which would prevail in an ordinary engine are set as the limits, and the temperature limits are determined by these

limiting pressures. It will result that while the same formulæ will apply as in the previous paragraphs, the quantities will differ because of the different values for the temperatures. Let the same initial temperature be chosen corresponding to 120 pounds pressure for steam, and the terminal pressure correspond to that in a condenser kept at 10 pounds above vacuum. Making the computations as before gives the following tabular result:

Vapor.	t_1	t_2	t_3	T_1	T_2	T_3
H_2O	341.60	194.00	104	802.26	654.66	564.66
C_2H_6O	286.34	158.34	104	747.00	619.00	564.66
CS_2	275.34	94.34	104	736.00	565.00	564.66
$CHCl_3$.........	295.34	125.34	104	756.00	586.00	564.66
NH_3......... $\Big\}$	66.34	-- 41.66	104	527.00	419.00	564.66
	66.34	39.34	39.34	527.00	500.00	500.00
$C_4H_{10}O$	231.34	141.34	104	692.00	602.00	564.66

Vapor.	p_1	p_2	p_3	L_1	L_2	C
H_2O	17408	1469	152	186905	20553	1.000
C_2H_6O	17408	1469	375	199649	20992	0.859
CS_2	17408	1469	1717	135730	15586	0.249
$CHCl_3$.........	17408	1469	920	114109	18786	0.242
NH_3......... $\Big\}$	17408	1469	33020	166700	17726	1.000
	17408	10406	10406	166700	10648	1.000
$C_4H_{10}O$	17408	1469	2530	146542	....	0.579

Vapor.	M	W	H	E	Lbs. Vapor necessary to produce 1 H. P. per Hour between 120 lbs. and 10 lbs. Pressure. W
H_2O	1.000	49084	237739	20.64%	11.44
C_2H_6O	1.193	57222	311249	18.38	30.48
CS_2	1.484	35109	182255	19.27	89.24
$CHCl_3$.........	1.135	37617	186359	20.13	78.15
NH_3 .... $\Big\}$	−0.204	−254827	−154483	165.00	....
	1.033	8579	306812	2.87	....
$C_4H_{10}O$........	4.877	29108	226734	12.84	90.30

The figure of 165 for the ammonia vapor is the result of
the unpractical assumption which makes the condenser pres-
sure greater than that of the boiler.   Hence a second engine
worked by the condenser will be required to operate the
engine imagined, and the figure belongs to the supplementary
engine.   The second ammonia line belongs to the supposi-
titious case of an engine operated with condensing water at
39° F.   This is not obtainable except by climatic accident or
by use of mechanical refrigeration.

It will therefore appear from the above:

(1) The rival for steam in theoretical efficiency is chloro-
form only, which requires nearly eight times the weight of
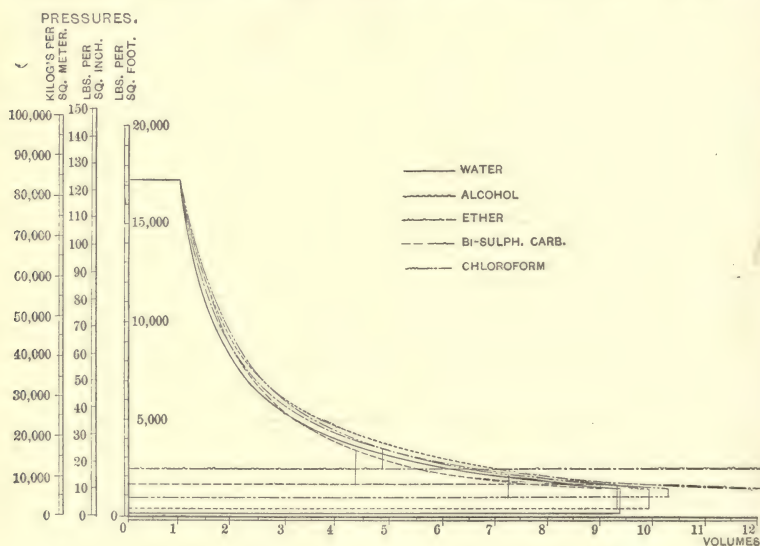
Fig. 177.

medium per horse-power per hour and must be operated in a
cylinder of one and one-quarter times the volume—both
tending to obliterate the small gain.

(2) This comparison is distinctly unfair to the vapors
other than steam, and makes them show at a disadvantage.

This is made clear by plotting the work-diagram for the vapors as in Fig. 177. Both ether and bisulphide of carbon are expanded below the line of back-pressure, or more than completely.

310. Effect on Efficiency of Volatile Vapors from Adjusting Final Pressure and Expansion Ratio.—If the back-pressure be fixed by a convenient adjustment of the condensing water in amount or temperature, a more favorable proportioning of the ratio of expansion may be made for certain of the vapors, so as to have their final pressures so adjusted to the initial pressure, which has been fixed as in the preceding case, that no loop occurs. This alters the final volume of the expanded vapor, or it may be effected by letting the final pressure be increased by such a quantity as will obliterate the loop. Keeping then the same initial pressures, but increasing the finals (p_2), the work will vary slightly, without a variation in the heat expenditure, and a change of thermal efficiency will follow. The values in metric units as presented by Gantt and Maury for this case give:

Medium.	p_1	p_2	W	H	E	Cylinder Volume.
Water............	84518	7043	267705	1330639	20.12	1
Alcohol	84518	7358	262614	1465207	17.92	1.035
Ether	84518	17190	162869	1094722	14.88	.845
Carbon bisulphide	84518	15844	162875	878729	18.53	.756
Chloroform	84518	10214	219200	1136084	19.29	.944

This table shows steam to have an advantage in thermal efficiency and in weight per horse-power per hour over all the vapors, but not in cylinder volume; it should also be remembered that the less efficiencies in this case as compared with that of the preceding paragraph are to be offset against the losses from the higher values for r in that case. Such losses

are probably the result of condensation at the higher ratios, and of friction.   Fig. 177 shows the change in diagram which results when the foregoing changes of condition are imposed and a more effective adjustment of condition is made.

311. Effect on Efficiency of Certain Vapors by an Increase in the Pressure Range.—Combining the results of the last paragraph with the computations of § 307, so that the temperature limits of a condensing engine are used without a complete expansion down to the condenser temperature with the attendant heat-losses from the cylinder-walls, another set of values are derived.   This may be done first with conditions assumed most favorable to the steam-engine, as in § 308, and then with respect to getting more favorable conditions for some or all of the other vapors, as in § 309.

The first alternative will be applied as in § 307, with initial temperatures of 172° C., final temperatures of 90° C., and a condenser temperature of 40° C. or 105° F., which is as low as can be conveniently secured for the year round in the temperate zone.   For this case the final results are:

Medium.	Work.	Heat.	Efficiency.	Kilograms per H. P. per Hour.	Cylinder Volume.
Water................	267705	1330639	20.12	5.19	1
Alcohol...............	525086	2894904	18.14	13.13	.488
Ether	751821	4447823	16.90	28.64	.232
Carbon disulphide......	360878	1798275	20.07	34.41	.314
Chloroform........... .	376517	2002139	18.81	42.06	.396

If the second alternative be chosen, and the conditions so adjusted to the qualities and properties of the vapors as to reproduce under this assumption the same back-pressure values as were used in § 309, then the final results are further improved for the volatile vapors as given in the following table:

Medium.	$\frac{p_1}{p_2}$	W	H	E	F	M
Water	$\frac{84518}{7043}$	267705	1330639	20.12	5.19	I
Alcohol.	$\frac{164600}{7358}$	622978	2894904	21.52	11.07	.853
Ether.	$\frac{318970}{17190}$	988752	4447823	22.23	21.78	.535
Bisulphide of carbon	$\frac{179230}{15844}$	428486	1798275	23.83	28.98	.550
Chloroform	$\frac{145720}{10214}$	462834	2002139	23.12	34.21	.761
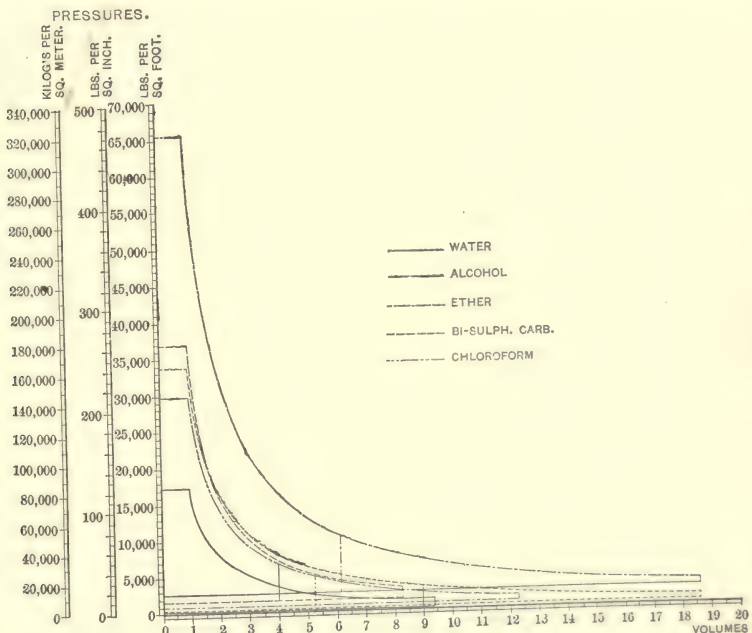
FIG. 178.

The indicator-diagram for this set of conditions is shown in Fig. 178 and presents the same peculiarities as to excessively high pressures for the other media than steam, to which

THEORETICAL ECONOMY OF VARIOUS VAPORS USED AS HEAT MEDIA IN A GIVEN ENGINE AT FIXED CUT-OFF, THE EXPANSION BEING SUFFICIENT TO CAUSE A FALL OF PRESSURE FROM 100 TO 14.7 LBS. PER SQUARE INCH. MEAN BACK-PRESSURE 14.7 LBS.

Substance used as a Heat Medium.	Specific Heat in Liquid Condition.	Temperature in Degrees Fahrenheit at pressures per square inch of		British Thermal Units necessary to be expended in order to generate 1 pound of vapor from "temp. of feed," equal to that given in column 4.	Efficiency or Fraction of Heat expended per col. 5 which can be realized as useful work in an engine.	Coal per hour per I.H.P., assuming the boiler to utilize 8000 heat-units per pound of coal—cylinder condensation neglected—in pounds.	Probable Coal per hour per H.P., including allowance of ⅓ for cylinder condensation for saturated vapors, and zero for superheated vapors—in pounds.
		100 lbs.	14.7 lbs.				
1	2	3	4	5	6	7	8
Steam............	1.00	328	212	1002	0.14	2.30	3.10
Ammonia, saturated ..	0.99	56	− 27	632	0.15	2.15	2.88
"　　superheated	0.99	282	8	940	0.13	2.48	2.48
Ether	0.55	215	95	195	0.15	2.15	2.88
Carbon bisulphide	0.24	248	115	163	0.17	1.99	2.68

reference has already been made. The table shows the direction in which efficiencies superior to that of steam are to be sought · but if this thoroughfare is closed, then the volatile vapors must be recommended for reasons other than their superior thermal efficiency or their economy in fuel per horse-power per hour.

In comment on the figures it should be added that con-densation during expansion or initial condensation (§ 224) would increase the weight of vapor, except where it was used in a superheated state.

If the steam is to be used without condensation, as in ordinary non-condensing engines expanding down to atmos-pheric pressure only, then the other media show to a little better advantage. The table on page 470 computed by Prof. D. S. Jacobus presents the data belonging to this assumption :

The conclusion safely to be drawn from this table is that if the comparison made with steam under a condition least favorable to its economy shows so little margin of advantage over the other vapors, there must be great advantages for them yet to be revealed if they are to be considered for large-scale work.

312. Usual Vapor Media. Their Disadvantages.— The foregoing paragraphs have enumerated the most fre-quently tried of the volatile vapor media. A list which should not omit some available ones would be perhaps hard to make. It should include, however:

Ammonia.......... (NH_3)	Gasoline.......... $(C_6H_{14}—C_7H_{16})$
Ether .... $(C_4H_{10}O)$	Carbonic acid gas.. (CO_2)
Carbon disulphide.. (CS_2)	Acetone........... (C_3H_6O)
Chloroform .... $(CHCl_3)$	Alcohol........... (C_2H_6O)
Naphtha .... $(C_6H_{14}-$ $C_8H_{18})$	Tetrachloride of carbon (CCl_4)

These have been tried or proposed separately, or in com-bination with steam, or even with each other.

The advantages to be urged for them are those consequent upon the lowered intensity of the source of heat to change

them into vapors with considerable tension. It therefore becomes possible to use a species of retort of small weight and bulk, within which a small weight of the medium can be injected at each stroke, to be at once volatilized and raised in tension, and to be worked in the working cylinder. This cannot well be done with steam, because the high temperature of such a " spray-boiler " for steam would result in its rapid corrosion and failure from deterioration of structure. There is therefore no reservoir of accumulated heat energy as in a steam-boiler, and its weight and bulk are avoided. Working pressure is promptly secured.

On the other hand, the disadvantages of these vapors, beside those already enumerated in the preceding calculations, are those which result from their comparison with steam.

(1) They have to be bought and paid for.

(2) They require to be operated as condensing engines, and demand a large body of cooling water.

(3) Some are inflammable, and their vapor mixed with air is explosive.

(4) Some are irrespirable, or produce unpleasant effects if inhaled, upon the human frame.

(5) Some have an odor, pungent or even offensive.

(6) Leakage must therefore be prevented at stuffing-boxes and elsewhere, by the use of a double-chambered construction, which is costly and troublesome.

(7) Some act corrosively upon metals, either alone or in combination with air or water.

An interesting computation under the second objection has been made which shows that with most of the vapors, to maintain a back-pressure of one atmosphere only, an amount of water is required which is one half of that which would maintain a 27-inch vacuum with steam. So that, in cities where condensing water is metered and taxed at a rate of $1.50 per thousand cubic feet required, the vapor-engine requires an expense for water equal to that for its fuel.

Unless the saving in fuel more than offsets this increased water cost, or unless the vapor-engine is to be run in places where condensing water is available without cost, the steam will show commercially at an advantage.

313. **Naphtha and Gasoline-engines.** — The use of naphtha in small launches instead of steam has been of considerable extent and interest in recent naval and pleasure-boat practice. Liquid naphtha stored in bulkhead tanks in the hull of the boat is used as fuel when mixed with air to heat a retort within which the pure naphtha liquid is vaporized, expanded, and brought to a high tension by the external heat. The vapor operates then in the engine, and is exhausted into a keel condenser, a coil of pipes outside the hull, and along the garboard strakes under the water. The machine starts by giving a few strokes to a hand air-pump to generate the naphtha-gas in the furnace, and by injecting by a second pump a small charge into the retort As soon as the pressure generated in the retort will drive the engine to compress the air for the furnace-gas, the compression of the air and the injection of liquid naphtha become automatic, and the process becomes continuous. The valves on the two naphtha circuits control pressure and therefore speed. The fuel-naphtha is the only supply which becomes exhausted, as the motor fluid operates in a closed circuit. The naphtha motor weighs but 100 pounds per horse-power in the smaller sizes, down to 75 pounds per horse power in the larger, which is to be offset against the combined weight of engine with its boiler and contents in the steam-yacht. The objection is the difficulty from having no reserve of energy in store on which to draw under unfavorable conditions of wind or waves. Fig. 179 shows a section of a naphtha-launch engine.

Yarrow & Co. state that the volume of vapor with naphtha in their launches produces ⅝ the volume which would be given by steam at atmospheric pressure. A 36-foot launch with 8-foot beam, weighing with its machinery and fuel-supply one

ton, will run 8 miles per hour in still water and can carry
fuel for 200 miles if necessary.   One and one half gallons per
hour, or one third gallon per horse-power per hour, will be
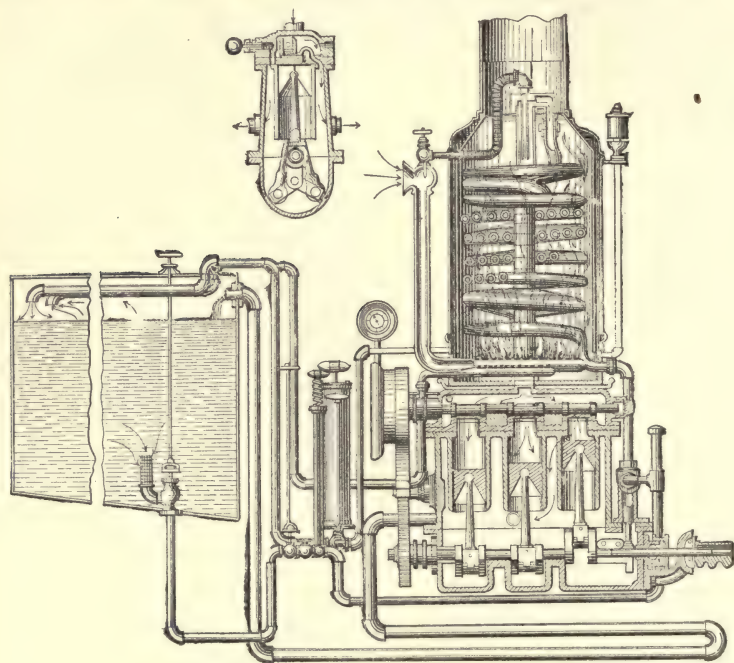
Fig. 179.

consumed in the furnace, and allowance for leakage and waste
must be added.

314 **Binary Vapor-engines.**—The experimenters with
vapors however, have usually preferred to retain the steam-
generator or boiler as a means for the convenient transfer of
heat to the volatile vapor they were to use as a motor fluid.
This principle has been used with the ether-engine, the
ammonia-engine, and the bisulphide-of-carbon engine.   The
scheme is illustrated in Fig. 180 as applied to a bisulphide
engine.   *S* is the ordinary steam-boiler   supplying heat
through the pipe *P* to the generator *AB*.   Condensed steam

from this generator is returned by a small feed-pump p. Within the generator AB is the closed chamber C holding the volatile liquid, ammonia, ether, or whatever. The heat of the steam-jacket vaporizes the volatile motor fluid, which passes through the pipe E to the motor-cylinder F. Exhausting thence into the surface condenser G with its provi-
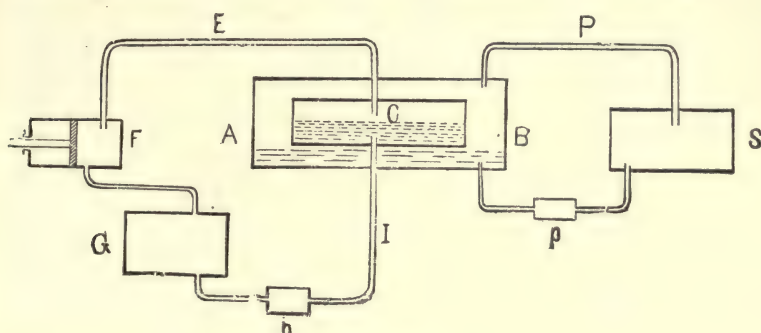
FIG. 180.

sions for circulating water through pipes or tubes within it to bring the motor-vapor back to a liquid, the latter is returned to C by the feed-pump h. The two circuits are closed and distinct from each other. No additional charge of either fluid is required except to replace leakage losses.

The other form of the binary plan is to add a vapor motor-cylinder to the steam motor-cylinder and have the two cylinders drive the shaft as in the compound steam-engine. The difference, however, is that the exhaust from the less volatile medium is made to pass through tubes or between thin plate surfaces on whose other side is the more volatile fluid. The tube or plate surface is a condenser for the hotter fluid, and the boiler for the more volatile one. The vapor is heated by the heat from the exhaust-steam of the first cylinder, and acquires tension to act to drive the piston in the vapor-cylinder. From this vapor-cylinder the vapor must pass to a condenser with abundance of cold circu-

lating water, to be from thence pumped back into the
generator to be used anew in circuit. This is the method
used by Du Trembley (1842) with ether, and by Ellis (1872)
with carbon bisulphide.

By the first plan no advantage is reaped from a saving of
furnace-losses in the boiler; by the second plan, if the steam-
engine were a wasteful one, some advantage might follow
from utilizing its wastes to evaporate the second fluid. If it
had no wastes, there should not be heat enough available
after exhaust to make the subsequent cylinder worth while,
and a loss in transfer of heat is unavoidable, since the im-
parting body must be hotter than the recipient. Nearly
3000 horse-power were aggregated in vessels on the Medi-
terranean service using Du Trembley's ether-engines before
the introduction of the compound steam-engine, but no care
could prevent escape of the vapor, and fire and other dis-
asters followed.

315. **Wellington Series Vapor Motor.**—In the spring of
1897 announcement was made that the late Mr. Arthur M.
Wellington of New York had been giving earnest study
previous to his death in 1895 to the problem of extending the
Du Trembley principle to the working of several motors in
series with successively more easily vaporizable media as the
lower end was neared. Starting with steam as having the
greatest absorptive capacity for heat in the first cylinder, the
exhaust was to pass into a special chamber of thin plate cells,
within which it should be cooled in the process of giving its
heat to the next medium, which was to be found on the other
side of the separating metal walls of the chamber. The
cellular chamber was therefore a boiler or heater for the
second medium, while a condenser for the first. The second
cylinder, operating with the second medium, exhausted into
a second condenser-heater for the third medium, and so on
to the last medium, which was condensed by cold water
when the descent down temperature had gone as far as it

was practicable to carry it. The fundamental idea is to work each medium within the limits of temperature range for which it is best adapted, and to transfer the heat from medium to medium by most effective contact methods of transfer. The principle is that of compounding by temperature but not by pressure, since the media are isolated in separate circuits, and do not act as back-pressures to those preceding it. The cylinder volumes will be proportioned for the pressure range belonging to the temperature range within which the medium works.
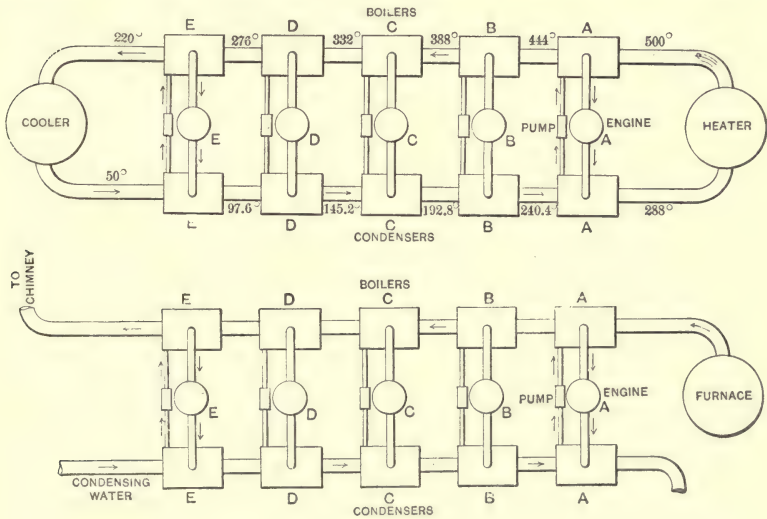
FIG. 194.

Fig. 194 will illustrate such a series engine both in the continuous system for one or a few media and in the successive system for a large number of media of varying properties.

Mr. Wellington also designed and patented efficient transferring appliances for the transfer of heat between the media, but his engine has never been built beyond an experimental stage. It is open to the practical disadvantages of the media which it uses, and it is further fair to call attention to the difficulties which the series principle involves.

The object of the series plan is analogous to that of compounding below the low-pressure cylinder in the steam-engine. To enable the designer to use a warmer temperature or a less volume of condensing water or a less bulk of condenser, the second medium is intercalated between the medium of higher heat potential and the condenser. The purpose is to diminish the inconveniences attaching to considerable transfers of heat in a short interval of time. On the other hand, as the *number* of transfers is increased, it must be remembered that there is a loss at each step in practice, because in order that the transfer may take place in reasonable time, or with reasonable surface and weights of metal, there has to be a difference of temperature between the hotter and cooler body. This is found in successive refrigerative processes to be conveniently allowed to reach 35°. Hence there is increased loss as the number of such transfers is increased. Furthermore, Fig. 181 shows the temperature-entropy diagram of a four-series engine in which the line 15–16 is fixed by the coolest available condensing water, and the area of heat rejected from the media above it gives the area of heat energy available for each successive medium. The shaded areas are separated by vertical spaces which represent the losses in transfer, and each rectangle is made wider as the succession moves down temperature, because with less value for T the factor ϕ should be greater to produce a given area. While each medium is assumed to act in a Carnot cycle, yet an enveloping line which would give a summation of the areas would inclose a less area than if the hottest medium had been used alone within the same temperature limits. The Carnot standard maximum efficiency assumes the entropy change to take place at the highest and lowest temperatures respectively. Hence while the series principle is defensible in theory, its practical difficulties as to size, weight, and cost of transfer apparatus usually more than offset the possible advantage.

An interesting combination of the series principle has been suggested wherein a gas-engine and a steam-engine are operated, so that the rejected heat from the gas-engine shall be used to make steam for the second cylinder. It is a question whether the gas-engine rejects heat enough to make this a practically utilizable combination when the steam cylinder is to be of any size. It would form an interesting computation for which there is no present opportunity.
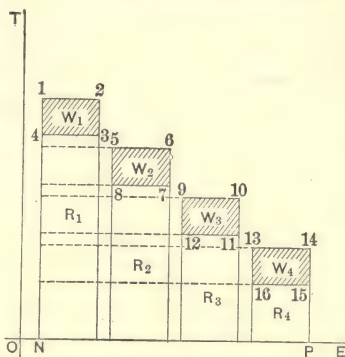
FIG. 181

The commercial obstacle to the series principle must not be overlooked. The bulk and weight of the series of cylinders and condensers makes an interest cost for the plant which causes a serious inroad into its possible fuel economy. The Wellington series was proposed as far back as 1872, in the Transactions of the Polytechnic Club of New York City, by Mr. T. D. Stetson.

316. Ammonia-vapor-engines. — Ammonia is employed usually like the bisulphide of carbon by steam vaporization in a closed circuit. It can, however, be used in a sort of absorption circuit, by causing the expanded ammonia-gas to meet a spray of ammoniacal water which, when cool, will absorb the ammonia-vapor. The latter can be separated as a gas on moderate heating, and can be condensed by cooling, or by cooling and compression, for use over again. This latter principle is rather along the line of the convenient storage of energy than that of the availability of ammonia as a heat medium. Where ammonia has been tried as a medium for naval uses it has been found to offer no advantages over steam.

317. Combined-vapor-engines. Æro-steam Engines. — In all the foregoing discussion of volatile heat media which

have to be purchased in the market the necessity for condensation is rigorous, and the consequent presence of the necessary cooling water. Where the facilities for condensation cannot be had it has been proposed to combine air and steam, and lift the efficiency of the latter by mixing with it the air, whose efficiency is calculated as greater than that of steam. This has been done in two general ways. Air under pressure from a pump has been forced into the steam-pipe, so that the cylinder-charge was of nearly equal proportions of steam and air (Mont-Storm's "Cloud" Engine, 1850–1881), or the more satisfactory plan may be followed of drawing hot air and gases from the flues and injecting them into the steam-space of the boiler (Warsop's and Wethered's methods). The effect of the hot air is to raise the temperature of the mixture without a corresponding rise in pressure, producing the equivalent of a superheating of the steam and lessening cylinder condensation (§§ 132 and 229). Some loss of heat in the chimney-gases is also prevented, but the presence of corrosives in the products of combustion with most fuels, made active by the reactions with hydrogen and at high temperatures, is likely to attack the metal of the engine.

The improved economy of modern steam-engines, using the higher steam-pressures and the principle of continuous expansion, have enabled as good results to be secured with steam alone as with the aero-steam combination as it has been used.

318. Storage of Energy in Liquefied Vapors.—The use, as motor fluids, of elastic media which can be compressed to liquids by mechanical means furnishes a compact and effective way of storing such energy for transport. Carbonic acid has been so used since 1823, and the use of what is called liquid air is now proposed. It must be remembered, however, that there is to be derived from the expansion of such mechanically compressed gases only the energy which was stored in them, less the heat withdrawn in cooling them to their state of stationary temperature in accord with their surroundings.

Unless reheated when used, their expansion and vaporization withdraws heat from their surroundings to an inconvenient extent, and in the case of carbonic acid or other gases which have a cost outside of the labor and fuel cost of compression, they are of significance only where expense is of less moment than some other object attained by their use. Compressed air at high tensions will usually meet these requirements more satisfactorily than the liquefied gases.

319. **Conclusion.**—It has been the intention that the reader and student shall derive from this chapter the conclusion that the vapor of water is the most effective of the various heat media when a balance is struck with respect to its advantages in competition with the others, unless certain extraordinary conditions are imposed as to weight or some other feature. It was intended that from the chapter on the gas- or oil-engine the conclusion should be derived that the use of these motors will extend more and more into fields now occupied by steam. Hence the gas- and the steam-engine are the two most important motive-power factors where available energy from falling water is not at hand by electrical transmission or otherwise.

With the completion of this chapter the subject of heat and heat-engines as laid out by the author for study and discussion is completed. It is intended that this treatment should be regarded only as the beginnings of the subject, and as leading to the mathematical discussion of thermodynamics as a deductive science from its differential equations. For such treatment the reader is referred to other treatises. The concluding chapters are convenient addenda having an obvious connection with the principal purpose of the book.

CHAPTER XXII.

MECHANICAL REFRIGERATION.

325. Introductory.—In the preceding chapters the object sought has been to liberate heat energy from fuel or other sources of heat, and to utilize this energy for industrial purposes in the form of mechanical work. The purpose of the present chapter is to consider the converse of this. Having a store or supply of heat energy either in the form of sensible or observable heat-units, or in the derived form of mechanical energy, it is sought to dispose of that energy in both forms, so that for equalization of heat condition a draft shall be made upon the store of heat energy in objects or places under their usual conditions. The result of this draft or lowering of the heat condition of an object or a place is what is generally recognized as the production of cold. It has been most aptly called a process of '' heat-pumping.'' By the expenditure of mechanical energy upon a convenient and well-chosen heat medium its condition as to temperature alone, or as to both temperature and entropy is lowered towards that represented by the absolute zero of temperature, just as the level of water in a vessel is lowered by expending through a pump the energy necessary to draw water out of it.

The process of expanding a heat medium in an engine from a high temperature T_1 to a lower one T_2 is a process of mechanical refrigeration. It is, however, only when this process is so located on the temperature scale that the drop down the scale begins at the normal temperatures and proceeds below the usual limits for the climate and the zone

where it takes place that the term refrigeration is properly used. This will suggest, however, the methods which are used in refrigerating cycles.

While in heat-engine practice the medium to be used is chosen with a view to its heat-carrying capacity, in refrigeration it will be chosen with a view to its willingness to surrender its heat energy to surrounding objects. The vapors have therefore signal significance as media in heat-pumping.

326. Analogy between the Heat-engine and the Ice-machine.—The foregoing analogy may be made more apparent by the use of an illustrative diagram (Fig. 185). Let the small cylinder on the left be the compressing or feed-water-
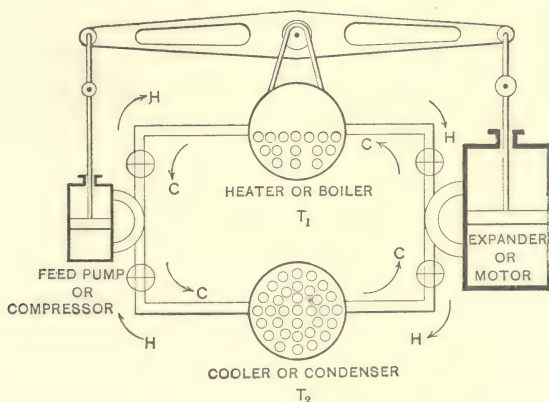
FIG. 185.

pumping cylinder of a heat-engine plant. If it be a steam-engine, the current will pass clockwise or upward to the left from C through P to B, which is the heater (or boiler). In the boiler its temperature is lifted from T_2 to T_1. Thence it passes to the working cylinder A, where it operates by its expansive energy to drive the piston, and is cooled by the supposedly adiabatic expansion to T_2, or the temperature in the cooler or condenser (C), which withdraws any heat which might be generated by the species of compression of the

medium on the return or exhaust stroke of the working piston. The compressing pump P draws from the cooler or condenser and completes the circuit.

When the same or a different medium is operated as a refrigerating circuit the compressing piston in P receives mechanical energy, compressing the medium and raising its temperature an amount proportional to the work expended in compressing it. The current moves contraclockwise, or from the compressor downward to the cooler C. Here the heat from the compression is withdrawn by a circulation of cooling water and is rejected with that cooling water, so that the compressed medium leaves the condenser at the temperature T_2. From the cooler the medium passes to the working or expansion cylinder A, where it does work against an exterior resistance (usually that of compressing the medium in P in part), and by such adiabatic expansion is cooled below the temperature T_2 of the condensing water, and to a degree far enough below the surrounding objects to be anxious to withdraw heat from them. From A the cooled and expanded medium passes to the organ B, which is a heater so far as the medium is concerned, while it appears as the place or material to be refrigerated. That is, it supplies its heat to bring up the medium to its own heat condition, and in so doing is cooled itself. From this refrigerating chamber or heater of the medium the medium passes back to the compressor and repeats the cycle.

It is apparent, therefore, that in the heat-engine there is heat supplied from without at the heater B in the beginning of the cycle, and withdrawn at the cooler C at the end; in the refrigerating cycle heat is withdrawn at the cooler C at the beginning of the cycle, and therefore surrounding objects must supply heat at the heater B to the medium at the end of the cycle to close it.

327. Refrigeration for Ice-making or for Cooling-chambers. Brines.—The heat withdrawn from the surroundings

of the heater *B* in the preceding paragraph, to warm the medium while being themselves cooled by such transfer, is most easily transferred from a liquid to the medium enclosed in coils of pipes. This liquid requires to be one which shall not be too liable itself to freeze by the cooling process, and is therefore a solution of such alkaline salts in water as shall have its solidifying point lowered. It is therefore generically known as " brine." A solution of Liverpool salt in well-water, of a degree of concentration such that it weighs 73 pounds per cubic foot or has a specific gravity of 1.17, will not sensibly thicken or congeal at zero Fahrenheit. American salt brines of the same gravity congeal at nearly 20° F. Chloride of calcium solution or chloride of magnesium can be used instead of brine. The latter remains fluid at 5° F.

When artificial ice is the object of the process there are two usual methods which are followed. The cooled brine surrounds thin metal cans within which is enclosed the dis-tilled water to be frozen in one system, which is therefore called the " can " method; or the cooled brine or ammonia itself in coils circulates between hollow plates, perhaps 10 feet by 14 feet in area, on the outside of which the ice forms in the water-tank. This latter is called the plate method. In the can system blocks weighing 300 pounds will be frozen in from 50 to 60 hours; in the plate system the ice forms 14 inches thick over their area in from 9 to 14 days. The can system is more usual for ice-venders, since the supply of ice-blocks is continuous, but the product of the plate system costs less, although the plant costs more.

When cold-storage chambers or cooling-chambers in fac-tories are to be kept at low temperatures for the sake of their contents, the cooled medium can be circulated in coils of pipes about the walls of the chambers, or a cooled brine may be similarly circulated. The present practice tends towards direct expansion except in special cases, as likely to maintain a more uniform mean temperature in the different parts of the

coils. Or, again, air which is cooled by passing over and through a brine coil may be circulated in the chamber by means of fans. Such air will deposit its moisture in the form of snow upon the metal of the coils, and a dry cool air only will reach the stored material, and condensation thereon will be lessened or avoided. Where no brine is used the system is called a " direct expansion " system.

328. Media for Use in Refrigerating-machines. Advantages and Disadvantages.—A refrigerating medium may properly be expected to meet as many as possible of the following requirements:

(1) If it is a vapor, it must be volatile at low temperatures, but at pressures not too far below that of atmosphere.

(2) At high temperatures it must not reach high pressures.

(3) It must be stable in its composition, so as not to alter by the frequent evaporations which it must undergo.

Whether a vapor or a permanent gas:

(4) It must have no effect on metals convenient for use in machine-making.

(5) It must be without effect on convenient lubricants such as will have to be used in cylinders.

(6) It must be non-inflammable if leakage occurs.

(7) It must be non-explosive.

(8) It should be without serious physiological effect on workers around the machinery.

(9) It should not be too costly to buy.

The discussion of vapors as heat media has presented several of the volatile vapors adapted for use as heat-carriers (§§ 107–109 and § 312). For the uses of the process of withdrawal of heat, while some of the same media will serve, there are other special ones which have been tried. The list includes:

(1) Water-vapor.

(2) Air.

(3) Ether alone.

(4) Ether mixed with SO_2 (Du Motay binary fluid).

(5) Anhydrous sulphurous acid or sulphur dioxide (SO_2).

(6) Mixture of No. 5 with carbonic acid (CO_2), "Pictet fluid."

(7) Ammonia (NH_3).

(8) Chymogene or other volatile derivatives of petroleum.

Ether is practically no longer in use, because the compressing cylinder has to have a volume six times that required for the sulphur dioxide and seventeen times that for an ammonia-machine. This follows from the density of the ether-vapor. It has also to be worked under less than atmospheric pressure, since its tension at 27° F. is 2 or 3 pounds per square inch, and the tendency of air to leak into the machine oxidizes the ether to a less volatile compound. Ether is also inflammable and acts on the lubricants to dissolve them. Such machines as used in India appear to have made 6 pounds of ice per pound of fuel consumed.

Sulphur dioxide is a liquid at 14° F., and at 60° to 65° F. has a tension of 3 to 4 atmospheres. It is without effect on grease used as a lubricant, and acts like one itself to keep metallic surfaces from contact. It is not inflammable and is stable, but is irrespirable. When moisture gets to it, the active acid is formed, which corrodes metals.

Carbonic acid (CO_2) used alone requires so high a range of pressures (800 pounds per square inch on the compressing side and 300 pounds on the suction side) that it is not practicable. The mixtures of Pictet and Du Motay have been displaced in America by the ammonia systems, on account of the convenience and cheapness of the medium and because capital has been attracted to invest itself in the manufacture of this type of machinery.

The petroleum-vapors are explosive and dangerous to use, and such machines are only experimental as yet.

The water-vapor machine uses a cheap and harmless medium, but the cylinders have to be enormous if the vapor operates in a compression cycle. In this case a vacuum-chamber

is required, in which by a pump a vacuum-pressure of about one tenth of a pound per square inch is maintained. A part of the water or brine injected into this chamber vaporizes, and the remainder is chilled in proportion to the latent heat extracted from it for the vaporizing of the first portion. The water-vapor thus produced may be handled in two ways. It may be passed into a surface condenser, the condensed liquid pumped out to waste, and the remaining vapor compressed from one tenth of a pound to one and one half pounds, to be worked as a superheated gas in a cycle similar to those to be discussed presently; or the absorption principle may be followed as in the ammonia-machines shortly to be discussed, so that the water-vapor from the vaporization in the vacuum-chamber is absorbed by sulphuric acid in a distilling apparatus. From the acid, which has a great affinity for water, the vapor is expelled by gentle heat, and after condensation is removed by the air-pump which maintains the vacuum. For ice-making conditions the volume of water-vapor would have to be 150 times that of ammonia. The machine is not in use in America to any extent.

The air- and the ammonia-machines are those of principal importance. The air-machine is principally used on shipboard, where pungent vapors from any leakage would be objectionable, and particularly in confined or ill-ventilated places.

329. Refrigerating-machines using Air as a Medium.— The discussion of the air-engine using air at ordinary temperature at admission and expanding it adiabatically (§ 260) should have made it clear that by the use of high grades of expansion the final temperature of the air will be very low. It will be recalled that

$$\left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \frac{T_2}{T_1},$$

in which for air $n = 1.41$, the ratio of the specific heats of

air. So that if the air be taken in at 68° F., and expanded so that the ratio $\frac{p_2}{p_1}$ has the values given in the following table, the final Fahrenheit temperatures will be those given in the table when calculated according to the formula:

$\frac{p_2}{p_1}$	Final Temperature Fahrenheit.	$\frac{p_2}{p_1}$	Final Temperature Fahrenheit.
2	− 28	9	− 180.5
3	− 75.9	10	− 188.9
4	− 106.6	11	− 196.3
5	− 128.7	12	− 202.9
6	− 145.7	13	− 208.7
7	− 159.4	14	− 214.1
8	− 170.9	15	− 218.9

The same truth is also apparent from a study of the table in § 182. These very low temperatures cannot be secured unless the incoming air be at a very great pressure, and moisture in the air and the conductivity of the cylinder metal limit the attainment of the theoretical figures. It is also difficult to cool the air in bulk to atmospheric temperature, so as to secure a cool admission.

The air-machine appears in two general forms. The closed-cycle machine, represented by the Allen Dense-air Ice-machine, and the open cycle, represented by the Bell-Coleman Machine of European practice. In the Allen machine, and others on the same principle, there are three cylinders taking hold upon a common crank-shaft. One is a steam-cylinder; number two is an air-compressing cylinder, taking its supply of air from the closed coil of the refrigerating-chamber, and compressing it to 200 pounds, or so, to the square inch. This compressed air is received into a cooling-tank or surface condenser, where the heat due to compression is removed so far as possible by circulating water, and the compressed air brought as nearly to the temperature of that water as is consistent with manageable bulk for the cooler.

From this cooling reservoir the compressed and cooled air
enters number three cylinder, in which it acts as in an expan-
sive air-engine, doing work upon the driving-shaft to relieve
the duty of the steam-cylinder. The exhaust from this air-
engine cylinder is at a low temperature, as indicated by the
table above, and is led into the cooling coils of the chamber
or brine, which latter delivers its heat to the pipes enclosing
the exhaust in the effort to equalize the differing tempera-
tures.

The open cycle of the Bell-Coleman type acts as in the
Joule air-engine reversed (§ 275). It draws air from the open
cooling-chamber at about atmospheric pressure; compresses
it into a cooling chamber with coils of pipes; expands it in a
second cylinder to the pressure of the open chamber, lower-
ing its temperature by such expansion, and finally exhausting
the cooled air into the refrigerating-chamber again. Fig.
186 shows a diagram of the organs of such a machine, and
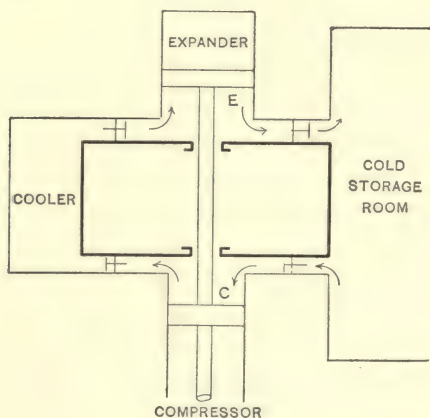
Fig. 186.

Fig. 187 the compression and expansion diagrams. The
closed curve *KadH* is the work done by the expansion-cylinder,
while *CHKb* is the work of the compressor, leaving the differ-

ence *abcd* to be provided for by the exterior work of the
steam-cylinder.

The standard tests of the performance of these two types
of machine (expressing them in heat-withdrawal equivalent
to melting pounds of ice at 32° F. into water at that tem-
perature, when the latent heat of fusion of ice is 142.2
B.T.U.) give 3 pounds of ice-melting capacity per pound
of fuel with the closed-cycle machine operating between 39
and 160 pounds pressure; and for the open cycle 3–4 pounds,
assuming the engine to run with 3 pounds of coal per horse-
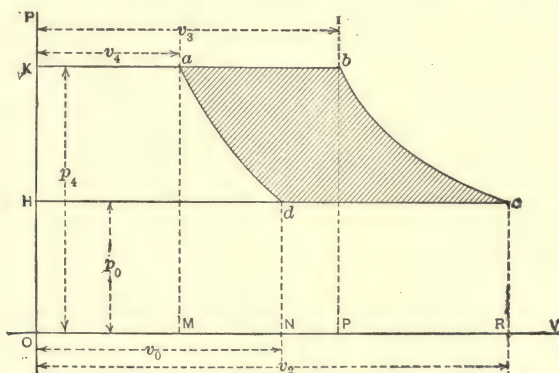
FIG. 187.

power per hour. The advantages which they offer are those
from the cheapness and harmless character of their medium;
their drawbacks proceed from the bulk and size of the
machine and the proportionate effect of the frictional resist-
ances; the effect of watery vapor in the air, causing snow in
the pipes and connections; and the effect of clearance·losses.
If high pressures are used, there is great difficulty in making
valves which will keep tight, and the heat from the com-
pressor-cylinder is sufficiently conducted to other parts of the
machine to cause a loss of effectiveness. With high tempera-
tures glycerine may be used as a lubricant.

**330. Ammonia Refrigerating-machines. Compression
Type.**—Reference has been made above (§ 328) to the

American prevalence of ammonia as a medium for producing cold mechanically. This results from the commercial accessibility of ammonia as a by-product in gas-making or coke-making, and the comparative ease with which anhydrous ammonia can be produced from ammonia-liquor by a fractional distillation process, since the ammonia-gas is more volatile than the steam-gas from water. The table on page 493 computed by the late Prof. De Volson Wood gives the properties of the saturated ammonia:

From this table the convenient adaptability of ammonia will be apparent on its physical side. It will remain a liquid under atmospheric pressure only when kept at the temperature of 30° F. below zero on that scale, and at the usual atmospheric temperatures in this climate of about 70° it will be kept a liquid only by maintaining it under a pressure of 115 pounds above the atmosphere. With reduction of pressure the liquid becomes a vapor, withdrawing from surrounding objects the heat necessary to change its state. This heat of vaporization at atmospheric pressure is 573 B.T.U., as compared with the 966 units required by water.

The volatile character of the anhydrous ammonia makes it unprofitable to use an expansion-cylinder in its cycle as is required with air, since the gas will expand of itself by simply allowing it to pass from a higher pressure vessel into another at lower tension and temperature through a regulating cock or valve, and in the change of state by such expansion the cooling effect is so much greater than that resulting from the exterior work done in an expansion cylinder, that the latter becomes negligible. Such separating-valve is called an "expansion-cock."

In using ammonia as a medium there are two principal systems, the compression system and the absorption system.

In operating a refrigerating plant on the compression system there is required a steam-engine operating an ammonia compressor; a cooler or surface condenser in whose

SATURATED AMMONIA.

Temperature.		Pressure, p. (Absolute.)		Heat of Vaporization. Thermal Units. r.	External Heat. Thermal Units. $\frac{pu}{778}$	Internal Heat. Thermal Units. $r_1 = r -$	Volume of Vapor per lb., cu. ft. v.	Volume of Liquid per lb., cu. ft. v_9.	Weight of a cu. ft. Pounds. $\frac{1}{v}$ of Vapor.
Degree F. t.	Absolute. T.	Lbs. per sq. ft.	Lbs. per sq. in.						
− 40	420.66	1540	10.69	579.67	48.25	531.42	24.38	.0234	.0411
− 35	425.66	1773.6	12.31	576.69	48.35	528.34	21.21	.0236	.0471
− 30	430.66	2035.8	14.13	573.69	48.85	524.84	18.67	.0237	.0535
− 25	435.66	2329.5	16.17	570.68	49.16	521.52	16.42	.0238	.0609
− 20	440.66	2656.4	18.45	567.67	49.44	518.23	14.48	.0240	.0690
− 15	445.66	3022.5	20.99	564.64	49.74	514.90	12.81	.0242	.0775
− 10	450.66	3428.0	23.77	561.61	50.05	511.56	11.36	.0243	.0880
− 5	455.66	3968.0	27.57	558.56	50.44	508.12	9.89	.0244	.1011
∓ 0	460.66	4373.5	30.37	555.50	51.38	504.12	9.14	.0246	.1094
+ 5	465.66	4920.5	34.17	552.43	50.84	501.59	8.04	.0247	.1243
+ 10	470.66	5522.2	38.55	549.35	51.13	498.22	7.20	.0249	.1381
+ 15	475.66	6182.4	42.93	546.26	51.33	494.93	6.46	.0250	.1547
+ 20	480.66	6905.3	47.95	543.15	51.65	491.50	5.82	.0252	.1721
+ 25	485.66	7695.2	53.43	540.03	51.81	488.22	5.24	.0253	.1908
+ 30	490.66	8556.4	59.41	536.92	52.02	484.90	4.73	.0254	.2111
+ 35	495.66	9493.9	65.93	533.78	52.22	481.56	4.28	.0256	.2336
+ 40	500.66	10512	73.00	530.63	52.42	478.21	3.88	.0257	.2577
+ 45	505.66	11616	80.66	527.47	52.62	474.77	3.53	.0260	.2832
+ 50	510.66	12811	88.96	524.30	52.82	471.44	3.21	.02601	.3115
+ 55	515.66	14102	97.93	521.12	53.01	468.01	2.93	.02603	.3412
+ 60	520.66	15494	107.60	517.93	53.21	464.76	2.67	.0265	.3745
+ 65	525.66	16994	118.03	515.33	53.40	461.82	2.45	.0266	.4081
+ 70	530.66	18606	129.21	511.52	53.67	457.95	2.24	.0268	.4664
+ 75	535.66	20339	141.25	508.29	53.76	454.70	2.05	.0270	.4878
+ 80	540.66	22192	154.11	504.66	53.96	450.75	1.89	.0272	.5291
+ 85	545.66	24172	167.86	501.81	54.15	447.75	1.74	.0273	.5747
+ 90	550.66	26295	182.80	498.11	54.28	443.70	1.61	.0274	.6211
+ 95	555.66	28566	198.37	495.29	54.41	440.95	1.48	.0276	.6756
+ 100	560.66	30980	215.14	491.50	54.54	437.35	1.36	.0277	.7353

coils the compressed ammonia may be cooled by the withdrawal of the heat due to the compression, which is made up
of the equivalent of the mechanical work of such compression

together with the heat necessary to produce vaporization, or the latent heat; the expansion-cock, through which the cooled and liquified ammonia may expand into a gas, withdrawing in so doing the heat from the brine which surrounds the coil or vessel into which the expansion takes place. Subsidiary circulating-pumps will then circulate the cooled brine in the chambers or water to be chilled. Fig. 188 illustrates the organs and connections for such a brine plant.

If the direct-expansion system is used, the expansion-valve at the bottom of the ammonia-receiver is replaced by a manifold from which groups of expansion-valves connect with the cooling-coils in the cold rooms at their lower ends. The ammonia-gas suction-pipe is connected to the upper or warmer ends of these cooling-coils by proper valves, and the circulation of the cooling medium is thus maintained.

In the actual operation of a plant the suction side of the compressor is connected to the low-tension side of the vessel or coil controlled by the expansion-cock, and a suction-pressure of from 5 to 20 pounds per square inch above atmosphere is maintained there by regulating the speed of the compressor and the opening of the cock. In starting with a new charge of anhydrous ammonia, it is introduced into this suction-chamber from the exterior vessel in which it has been received from the chemical manufacturers. Coming into the suction-chamber or coil at atmospheric temperature, the liquid ammonia volatilizes under the reduced pressure, and the proportion to be volatilized will be determined by the temperatures surrounding the coil or suction-chamber. Usually 10 per cent so volatilizes, cooling the entire mass down to the temperature of ebullition proper to the suction-pressure. Then the remainder of the liquid ammonia (or 90 per cent) volatilizes by the withdrawal of heat from the surrounding coil and brine, and passes as a gas to the compressor-cylinder.

The compressing stroke raises the ammonia-pressure to perhaps 150 pounds and 70° F., and expels the gas to the
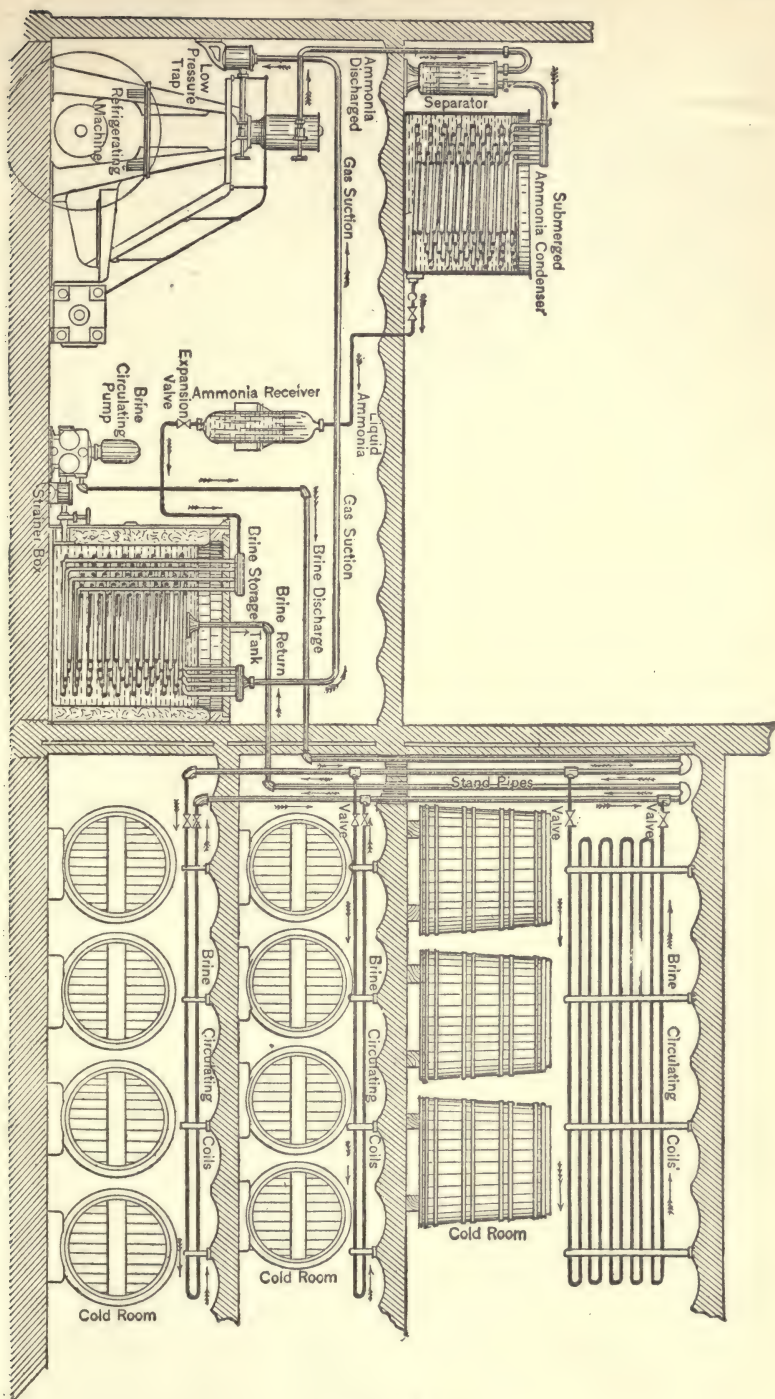
FIG. 188.

cooler or condenser, where the heat is withdrawn by circulation of water and submergence in it, and the gas condensed to a liquid again upon the high-pressure side of the expansion-cock in the ammonia-receiver. Upon opening the latter, the cycle of the charging process is repeated, and the process goes on thereafter continuously. The amount of ammonia charge in a machine will vary with the amount of piping in the circuit. A usual allowance averages 0.3 of a pound per running foot. Leakage or wastage ought not to amount to 100 pounds a year in a machine of 50 to 75 tons of ice capacity per 24 hours.

It will be apparent that the temperature of the cooling brine will be determined by the boiling-point of the ammonia, and that this will be determined by the pressure of the suction side of the compressor. The brine is usually 6° colder than the space it cools, and about as much warmer than the vaporizing ammonia. When, therefore, a temperature as low as 0° F. is required, as in storage of fish, the suction-pressure is kept down to 5 pounds; for brewery work, where storage temperatures of 34° F. are low enough, a suction-pressure of 28 pounds will suffice. Cold-storage chambers for fresh meat can be maintained at 25° F. with 24 pounds suction-pressure.

331. Wet or Cold and Dry or Hot Systems of Ammonia-compression.—If the regulation of the supply of ammonia to the cooler on the suction side of the compressor is so adjusted to the pressure that vaporization is complete, the machine is said to operate dry, or to belong to the dry-compression system. If, on the other hand, some liquid ammonia is allowed to remain unvaporized, or if some liquid ammonia is injected into the space filled with vapor, the system will be known as the wet or cold compression system. Prof. Linde introduced the first wet system, which is often known by his name, and Mr. George Richmond the second or injection method. The difference in the wet and dry systems follows from the presence of the liquid ammonia in the com-

pressing cylinder. If sufficient liquid is brought into the compresser, the temperature cannot there be raised higher than the boiling-point corresponding to the highest temperature and pressure of the compression; while with the dry saturated gas the temperature may go much higher than the limit set by the other system. More circulating water, however, will be required in the latter case because the compression-cylinder must be water-jacketed, which is not necessary in the cold system. Tests seem to show the two methods to give about equal results in economy, the reactions of the hot cylinder-walls neutralizing the theoretical advantage of the wet system.

332. Ammonia Refrigerating-machines. Absorption Type.—In the absorption system of ammonia refrigerating-machines advantage is taken of the property of water or of a weak ammonia hydrate, whereby it shows a strong avidity to dissolve in itself the dry ammonia-gas. At 59° F. water will absorb 727 times its volume of ammonia-vapor. Hence if a pipe leading to such an absorber be connected to the lower tension or cool side of the expansion-cock, the anhydrous vapor will be drawn off through it to unite with the weak liquor in the absorber at a rate comparable to that at which the suction-stroke of the ammonia-compressor acted in the compression system. The liquor in the absorber is drawn off by an ammonia-pump and fed into a still, or chamber, within which a coil of hot steam-pipes will vaporize the ammonia-gas, and crack the volatile ammonia from the less volatile water. The ammonia-gas may be at the same pressure and temperature as it would have had under the compression system, and is led into a condenser, where it becomes liquid anhydrous ammonia, to be worked through an expansion-cock as in the other system. Hence it will appear that the difference in the two systems results from a replacing in the absorption system of the compression-cylinder and its driving-engine by a vessel called the absorber with an ammonia-

liquor pump and a steam still. The cooling condenser, the expansion-cock, the brine-cooling appliance and its pumps will be identical for both systems. It usually follows that the absorption plant will be somewhat cheaper to install, and in many places it is convenient to avoid the running of the large compressor and its attendant expense. It is often convenient also to be able to use the commercial ammonia hydrate of 62 per cent water and 38 per cent ammonia with a specific gravity of .880. Fig. 189 shows a type of absorption plant with the essential organs.

ABSORPTION SYSTEM.
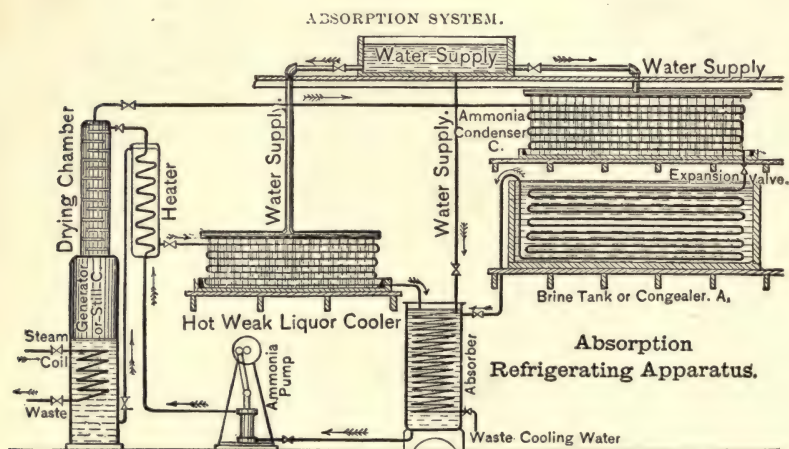
Fig. 189.

333. Refrigerating-machines on Pictet System. Carbonic-acid Machines.—Previous to the more extensive introduction of the ammonia-machines into America, the Pictet machine was the most extensively employed. It is still in limited use, and is also of importance in Europe. The medium at first was pure sulphur dioxide (SO_2), but is now more usually the so-called Pictet fluid of 97 per cent of SO_2 and 3 per cent of CO_2. The dilution with carbonic acid enables a temperature 14° F. lower to be secured at atmo-

spheric pressure than can be attained with the pure sulphur dioxide. The Pictet machine requires a little greater bulk for its compressing cylinder than the ammonia-machine, but otherwise they should be of equal theoretical efficiency.

The pure carbonic-acid machines require to work through a range of pressures higher in the scale than the foregoing, and difficulties from leakage and from the tightness of valves have stood in their way. To operate between 5° F. and 64° limits of temperature, the suction-pressure has to be 300 pounds per square inch, and the compression over 800 pounds. The compression-cylinder will be of one quarter the volume of the equivalent ammonia-machine, and this fact has given interest to the use of the machine for yachts and similar marine conditions, where room and weight are of paramount importance. The carbonic acid is more effectively worked in an expansion-cylinder rather than through an expansion-cock.
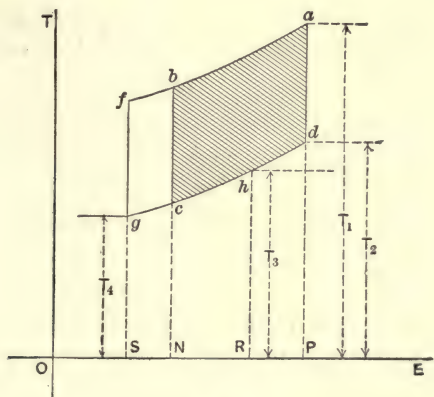
Fig. 190.

334. Temperature-entropy Diagram of Refrigeration Cycle.—The discussions which have preceded should have made easy the graphic representation of the useful effect by the use of the temperature-entropy diagram (Fig. 190). The

expansion of a permanent gas medium in an expansion-cylinder is practically adiabatic. Starting, therefore, from the upper right-hand corner and moving towards the left, a line *ab* is described which is a curve of constant pressure and decreasing temperature caused by the cooling by the condenser-water to the point *b*. Here adiabatic expansion down temperature takes place through the expansion-cylinder to the point *c*. The brine heats the medium and increases its entropy and temperature along the constant volume line *cd*, and at *d* the compression raises the temperature without change of entropy from *d* to *a*. If the medium used is liquefiable gas, the changes of entropy occur at constant temperature and the lines *ab* and *cd* are horizontal, giving the Carnot cycle diagram of Fig. 80.

In direct-refrigerating or open systems (§ 327), using air for example, where the cooled medium is exhausted into the space to be cooled, it will usually happen that the line *cd* will not be a continuous one, since the refrigerating-chamber will not raise the temperature of the medium to that corresponding to *d*. If *h*, for example, represents some such temperature level T_1, then the line through *h* represents a loss between the medium and the compressing cylinder due to the warm metal walls of the latter, and for which no refrigerating effect is produced. If the cylinder could be kept cool enough to have adiabatic compression on the second or succeeding strokes begin at *h* and describe an adiabatic vertical through *h*, it would appear that by successive withdrawals from the continuously cooling refrigerator a continuous lowering of T_2 could be made to follow. The practical limit is soon reached, however, from the impossibility of carrying the cylinder-metal temperature down as the process goes on.

If, however, the object of prime interest be the securing of a low temperature, the principle of the regenerator may be introduced which was discussed for the reverse process under Hot-air Engines (§ 268). The regenerator in a refrigerating-

machine is usually called an interchanger, and is applied to restore the heat to the medium which is called for by the gap between h and d. This lowering of the interchanger temperature may be used to cool the medium below the first
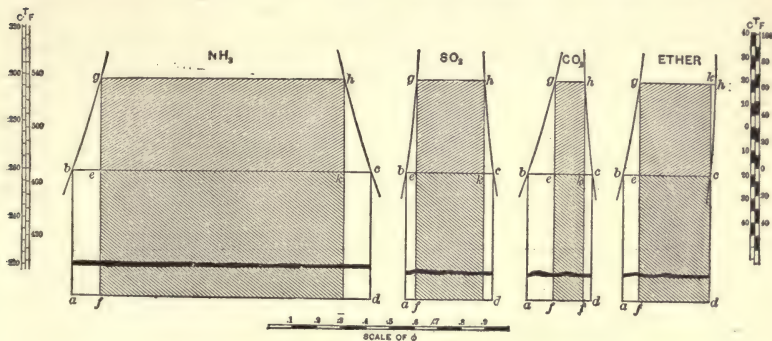
FIG. 191.

temperature corresponding to b, so that a lower entropy ordinate is reached down which the adiabatic expansion occurs (such as fg, for example). This lower range can be
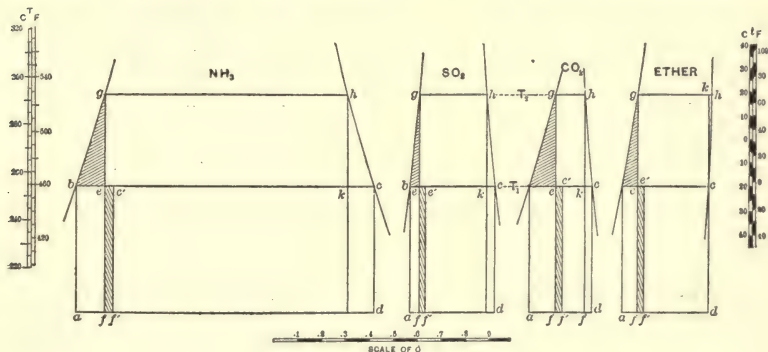
FIG. 192.

utilized on the next circuit to carry the cycle still lower, the effect being cumulative as long as the expansion-cylinder can have its temperature lowered at the same pace. This latter imposes the practical limit.

Figs. 191 to 193, reduced from a very complete paper by Mr. George Richmond (see Appendix), present the temperature-entropy diagrams of ammonia, SO_2CO_2, and ether when these media are operated by using a compression and an expansion cylinder through the Carnot cycle. Fig. 192 shows the diagrams which result when the expansion-cylinder is omitted, and the media are operated with some of the liquid present

TABULAR DATA FOR CONSTRUCTION OF DIAGRAM FIG. 191.

(1 lb. of each.)

	T_1	T_2	L_1	L_2	$\phi_1 = \frac{L_1}{T_1}$	$\phi_2 = \frac{L_2}{T_2}$
Ammonia..............	....	....	580.56	543.03	1.2501	1.0367
Sulphur dioxide	464.4	523.8	170.82	156.76	.3678	.2993
Carbonic acid........	$t_1 =$	$t_2 =$	121.50	67.93	.2616	.1297
Ether...............	....	....	170.99	166.14	.3682	.3172
Represented on diagram by...........	ab	fg	rectg. bd	rectg. hf	bc	gh

	$h = \int_{T_1}^{T_2} cdt$	$\Delta\tau = \int_{T_1}^{T_2}\frac{cdt}{T}$	$f = \Delta\phi + \Delta\tau$	C	$T_3 = T_{2\epsilon}\frac{f}{C_p}$
Ammonia............	60.10	.1216	− .0918	.5084	627.47
Sulphur dioxide	21.59	.0437	− .0248	.1544	156.00
Carbonic acid........	56.72	.1148	− .0171	.2	570.5
Ether	31.47	.0698	+ .0188	.475	....
Represented on diagram by	area $abgf$	be	kc		

so as to prevent superheating and so as to give the wet or cold system. Additional work is required as represented by the area bge, but the heat removed is reduced by the same amount, because the heat is not removed from the liquid as in the other set of conditions, and reducing the area of refrigeration by the area ef', equivalent to bge. When the

media are used in the dry system there will be a rise in temperature and entropy after the point h is reached, and additional work will be done as represented by the area of $khjc$ (Fig. 193). Refrigeration is also increased by the area of the rectangle kd.

$$e = \frac{kf - ef' + kd}{gk + bge + khjc}$$
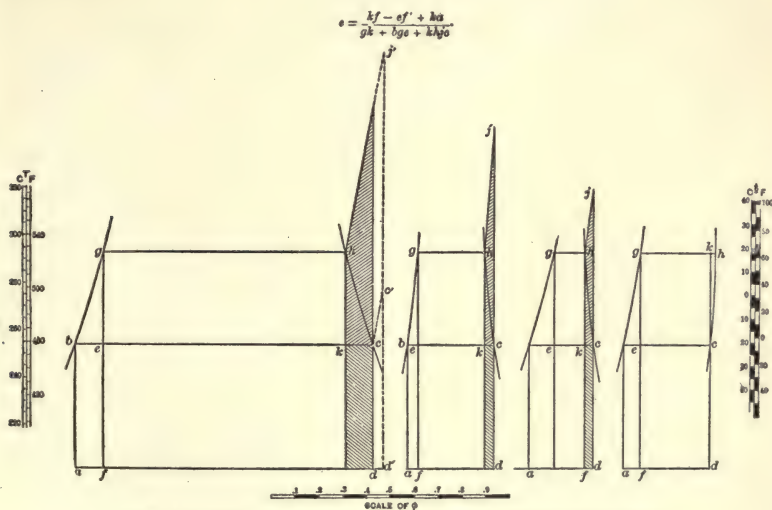
CASE IV.—DOTTED LINES, FIG. 90 : PRACTICAL CYCLE.—SUPERHEATED AGENT.

$$e = \frac{kf - ef + kd + dcc'd'}{gk + bge + khjc + cjj'c'}$$

FIG. 193.

335. Efficiency in a Refrigerating Cycle.—The temperature-entropy diagram also makes apparent to the eye the meaning of the expression for the efficiency in a refrigerating cycle. In the heat-engine cycle or the direct cycle under the Carnot hypothesis, the high temperature of the source of heat is the starting-point, and the heat energy flows downward through the heat-engine, which utilizes it, and that which is not available for work is transferred to the refrigerator or condenser. The efficiency is then the fraction whose denominator is the maximum heat energy available, and whose numerator is that part of it which is transformed into work.

The heat rejected will measure the excess of the denominator as compared with the numerator.

In the refrigerating cycle, which is the reverse of the direct cycle, the problem is to withdraw heat energy from the refrigerator and transfer it to the heater. Universal experience and the dictum of the second law of thermodynamics indicate that this can only be done by the expenditure of mechanical energy, or by a process analogous to pumping. The object will therefore be to remove the greatest quantity of heat energy by the expenditure of the least mechanical energy possible to accomplish this result. Efficiency of such heat-pumping will therefore be measured by a fraction whose denominator will be the mechanical work required, and whose numerator will be the entire amount of heat pumped out of the refrigerator. The difference between the total heat discharged into the condenser, and the heat pumped out of the refrigerator will be the equivalent of the work expended, and it will be obvious that the less heat converted into work, the more cooling is done and the more efficient the apparatus as a refrigerating device.

Referring then to the temperature-entropy diagrams, it will be apparent that the total heat energy of the cycle will be the product of the upper temperature value into the entropy range. Calling this upper value T_2 and the entropy range ϕ, we have

Heat energy obtained from refrigerator $= \phi T_2$.

The work of the compressor and expander is to take the medium used and operate with it between the larger value T_2 and the lower value T_1. The work done will therefore be $778\phi(T_2 - T_1)$ in foot-pounds or $\phi(T_2 - T_1)$ in heat-units.

Hence the efficiency by definition will be

$$\text{Efficiency} \quad \frac{\text{Energy obtained}}{\text{Energy expended}} = \frac{\phi T_2}{\phi(T_2 - T_1)} = \frac{T_2}{T_2 - T_1},$$

which is obviously an expression in the form of the reciprocal of the efficiency of a heat-engine. The peculiarity of this expression is that its value is greater than unity. This is due to the peculiar negative unit which it presupposes when the object is the pumping down of heat energy as a positive operation. Where the energy obtained is a part of the energy expended, as is the usual case, this apparent difficulty is avoided. For this reason many engineers prefer to use the relation between the heat actually withdrawn *as heat* to the work expended in the heat-pumping and call this the efficiency. That is,

$$\text{Efficiency} = \frac{\text{Cooling effect}}{\text{Work required to produce it}}.$$

This may become unity when equal areas of the temperature-entropy diagram are formed by the position of the line through T_1. As the work area grows less than one half the total heat-energy area, the efficiency is greater and transcends unity.

In either use, however, an important pair of conclusions is to be drawn:

1st. The efficiency increases as the temperature of the refrigerating-room increases;

2d. The efficiency increases as the temperature of the condenser or cooler is lowered.

The practical use of the efficiency results from the fact that it gives the number of thermal units removed per 778 foot-pounds expended. If the latent heat of fusion of ice be called 142.2 B.T.U. per pound, there will be required per ton of 2000 pounds made per 24 hours an amount of heat represented by

$$142.2 \times 2000 = 284400.0 \text{ heat-units.}$$

Each horse-power per 24 hours will be represented by

$$\frac{33000 \times 60 \times 24}{778} = 61079 \text{ heat-units.}$$

Hence the number of tons of ice made from water at 32° F. per horse-power per day of 24 hours will be

$$E \times \frac{61079}{284400} = \frac{\text{Heat}}{\text{Work}} \times .214 = E \times .214,$$

in which the heat and work values can be scaled from the temperature-entropy diagram.

336. **Refrigeration by a Series Process. The Step-by-step Method.**—It is obviously possible in refrigeration also to make use of a succession of media for producing a low degree of temperature just as in the series heat-engine it was sought to widen the range of pressures within a given temperature range (§ 315). The advantage secured is that by using a series each agent selected may be best adapted to the particular range of temperature through which it is used.

If the Carnot cycle be assumed for each medium in the

Richmond
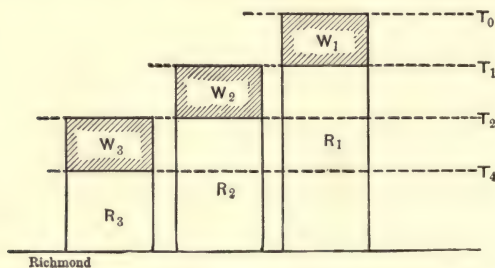
FIG. 195.

series, the operation as presented by the temperature-entropy diagram will be conducted in a series of steps as shown in Fig. 195. The medium adapted for the top of the range may act upon the material to be cooled by lowering its temperature from T_0 to T_1 by the adiabatic expansion of the medium down temperature through this range. The body to be cooled may then come into the second machine operating with a medium adapted for the lower range between T_2

and T_1, and by a similar expansion process may have its temperature lowered to T_2, and a third step with a third medium may drop its final temperature to T_4. Using again the analogy of heat-pumping, the action may be compared to a discharge of heat energy by each machine of the series into the suction or refrigerator of that above it, so that the work is done in three (or more) steps instead of doing it by the one step which might be difficult and even impossible. This method is used in liquefying some of the more difficult gases, and when used alone or in combination with free expansion methods can produce the lowest known limits of temperature.

337. Design of a Refrigerating-machine.—It would be manifestly impossible within the limits imposed by this discussion to refer exhaustively to all possible machines and combinations. The treatment of the air-compressor (Chapter XIII) and of the air-engine (Chapter XVIII) furnish all necessary formulæ for an air-machine, or one using a compression and an expansion cylinder. The heat to be withdrawn from the cold-storage room will be for each pound of air passing through the refrigerating-machine

$$C_p(t_1 - t_4),$$

when t_1 corresponds to p_1, the pressure at which the air enters the compression-cylinder, and t_4, corresponding to p_4, is the temperature in the cold chamber. If the amount of heat to be removed is given in units as well as the range, then the heat withdrawn will be

$$Q = MC_p(t_1 - t_4),$$

when M is the number of units of weight of air to be taken care of per minute. Q is usually given in terms of pounds of ice made or melted in a given time. If, for example, it be 400 pounds of ice per hour, then

$$M = \frac{Q}{C_p(t_1 - t_4)} = \frac{142.2 \times 400}{60 \times 0.2375(t_1 - t_4)}.$$

The work of compressing M units without account of clearance will be

$$W_c = M p_1 v_1 \frac{n}{n-1} \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\},$$

as shown in § 181, which may be written

$$W_c = \frac{MC_p}{A}(t_2 - t_1)$$

since

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \quad \text{and} \quad C_p - C_v = AR = C_p \frac{n-1}{n}.$$

The work in the expansion-cylinder (§ 254) will in like manner be

$$W_e = \frac{MC_p}{A}(t_3 - t_4),$$

when t_3 corresponds to the p_3 at the point of cut-off in that cylinder, and t_4 corresponds to the final pressure supposed to be that in the chamber when the expansion is complete, and the machine exhausts into the cooled chamber direct. Then the steam-cylinder must furnish a power enough greater than

$$W = W_c - W_e$$

to overcome its own friction and the friction of both the compressing and expanding cylinders, and the fly-wheel shaft and engine mechanisms. W being expressed in foot-pounds per minute and properly increased,

$$\text{H.P.} = \frac{W'}{33000}.$$

The cooling water required must be sufficient to cool the weight of air per minute from the compression temperature t_2

to that at which the air enters the expansion-cylinder at t_3; this will mean that a quantity of heat must be taken care of which is represented by

$$Q_w = MC_p(t_2 - t_3).$$

If the cooling water comes into the cooler at an initial temperature t_i, and leaves it with a final temperature t_f, the heat of the liquid for each pound is changed from q_i to q_f, so that a weight of water per minute H is required such that

$$H(q_f - q_i) = MC_p(t_2 - t_3),$$

or

$$H = \frac{MC_p(t_2 - t_3)}{q_f - q_i}.$$

With a compression-machine, operating in a closed cycle, and using a brine as a transfer agent, the problem usually comes in the form which calls for the dimensions of a machine having a certain ice-making or ice-melting effect per hour or per 24 hours. The weight of medium is then determined by the condition that each pound of it must withdraw from the refrigerant brine the heat required to change the liquid at t_1 in the condenser with a heat of liquid in each pound, represented by q_2, into saturated vapor with a temperature t_2 in the vaporizer, where the total heat of vaporization for the pound will be represented by λ_2. Hence the heat withdrawn per minute per pound will be

$$Q = \lambda_2 - Q_1.$$

When the heat to be withdrawn is given in the form of ice made per hour, then the weight of medium (M) becomes per minute

$$\frac{142.2 \times \text{weight of ice per hour}}{60} = Q_1 = M(\lambda_2 - Q_1).$$

The compressing work being assumed to be adiabatic, the temperature (T_s) of the superheated vapor leaving the cylinder to go to the cooling-condenser will be

$$\frac{T_s}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}.$$

Hence a quantity of cooling water in the condenser will be required which will be represented by

$$Q_w = [MC_p(t_s - t_1) + r_1],$$

in which C_p is the specific heat of the superheated vapor at constant pressure, and r_1 is the heat of vaporization (equal to $\lambda - q$, §§ 125, 127) of the medium at the pressure p_1 of the condenser. To find the weight of cooling water per minute, where the initial and final temperatures for it are given, the method above given for the air-machine may be repeated.

The net horse-power of the steam-cylinder which drives the compressor must be 778 times the heat-units to be disposed of. Hence

$$\text{H.P.} = \frac{778 \times M[C_p(t_2 - t_1) + \lambda_1 - \lambda_2]}{33000},$$

in which λ_1 and λ_2 are the respective total heats of vaporization at the pressures and temperatures in the condenser and the vaporizer. To this first or net result must be added the percentage allowance for the loss in the compressor, and the further allowance for the friction and other losses of the engine and compressor mechanisms.

The compressor-cylinder volume will be found by taking from tables the volume of one pound or unit weight of the medium. If this be called v_2, then the volume of M pounds (Mv_2) will equal the required volume to be swept through per minute; or if the compressor makes N strokes per minute,

$$Mv_2 \times \frac{1}{N} = \text{volume for one stroke.}$$

Corrections for clearance loss must be made as discussed in §§ 186, 187.

338. Performance of Refrigerating-machines. — The most complete résumé of the theory and performance of refrigerating-machines is to be found in the American issue of Ledoux's classic treatise on Ice-making Machines (see Appendix). From the tables there published it appears that on the assumption of an evaporation in a good boiler of 11.1 pounds of water per pound of combustible from and at 212° in an absorption-machine, or on the assumption of 3 pounds of coal per horse-power per hour with a compression-machine, an actual ice-melting capacity per pound of coal should be attained as follows:

With a brine temperature entering the refrigerated space of		The ice-melting capacity in pounds ranges between	
37° Fahr................	33 and 46	Average	39 +
28°	....	"	24
23°	....	16 and 33	"	24 +
8°	....	17 and 22	
− 5°	....	10 and 16	"	13 +

339. Freezing-mixtures. Some Low Temperatures. Liquid Air.—A limited degree of artificial cold results from the heat required for chemical reaction in certain mixtures of salts or salts and acids. Such freezing-mixtures are:

	When mixed by weight.	Reduce Fahr. temp. from	
Sodic sulphate, $NaSO_4$	8 parts		
Hydrochloric acid, HCl	5 "	50° to	2°
Sodic chloride (common salt), $NaCl$	1 "		
Snow or pounded ice, H_2O.................	2 "	50° to	0°
Sodic sulphate, $NaSO_4$.....................	3 "		
Dilute nitric acid, HNO_3..................	2 "	50° to − 2°	
Sodic sulphate, $NaSO_4$.....................	6 "		
Ammonic nitrate, $NHNO_3$.................	5 "	50° to −14°	
Dilute nitric acid, HNO_3..................	4 "		
Sodic phosphate, $Na(PO_5)_2$................	9 "		
Dilute nitric acid, HNO_3..................	4 "	50° to −20°	

The lowest ranges of temperature secured by the physicist

have been in the liquefaction of gases by compression and cooling at the high pressure, with subsequent lowering of pressure by expansion. Faraday of England (1820–1860), Natterer of Vienna (1854), Cailletet and Pictet of Paris (1878), Dewar of London (1893), and Chas. E. Tripler of New York are names identified with these researches and results. Some of these low temperatures or the boiling-points of liquid gases are:

For carbonic acid, CO_2 . . . . — 220° F.
For oxygen, O . . . . — 300° F.
For nitrogen, N. . . . . — 315° to 320° F.
For air, O + N. . . . . — 312° F.
For hydrogen, H. . . . . — 418° to 440° F.

Liquid Air is a product of considerable scientific interest by reason of the accessibility and cheapness of the agent. It is manufactured in 1899 by mechanical compression in three stages with intercooling (§ 189). The first cylinder ($10\frac{1}{2}'' \times 16''$) raises it from atmosphere to 65 pounds pressure; the second ($6\frac{5}{8}'' \times 16''$) from 65 to 400; and the third ($2\frac{5}{8}'' \times 16''$) from 400 to 2000 or 2500 in the usual process of to-day. The cooling is done by ordinary water circulation in coils up to the stage of storage in proper reservoirs of the cleansed high-pressure air. The air is then allowed to expand down to nearly atmospheric pressure through a specially designed expansion-valve into an expansion-chamber wherein the high-pressure air is circulating in strong coils. The lowering of temperature at that pressure combined with the regenerating effect produced by the contact with the coils is sufficient to transform the air to a liquid combination of its two elements, and it can be drawn off in drops through an outlet-valve. Such an apparatus as described with a steam-cylinder $16'' \times 16''$ at 85 to 90 pounds boiler-pressure makes from 2 to 3 gallons of liquid air per hour at a

cost stated to range from twenty cents to one dollar per gallon.

Liquid air of course vaporizes again, withdrawing heat from the air and surrounding objects to the amount required to restore it. It is in this way that it can be used as a source of motive power. The difficulty is to control the evaporation within convenient limits of pressure-resisting power of containing vessels. Nor should it be forgotten that in withdrawing heat from the air or surrounding objects this withdrawal must be compensated elsewhere, or else the heat-level of the world is lowered towards a level of cold for which the human organism is unfitted.

CHAPTER XXIII.

THE INJECTOR.

340. Introductory. — The discussion of the dynamic theories belonging to heat-engine practice would not be complete without a reference to the injector as an appliance for introducing the heat medium into the boiler or generator against the pressure in the latter. The injector differs from a feed-pump for this same purpose, because the latter is based upon easily understood principles of pressure and resistance, and a greater volume of steam displaces a less volume of water where the head of steam-pressure and the resistance-head of the water-cylinder are the same. The injector, on the other hand, depends upon a direct conversion of heat energy into dynamic energy, and by a process not so obvious or plain as in the case of the steam-pump.

341. The Injector Defined. The Ejector. —The injector may be defined as an appliance or apparatus whereby a jet of steam moving at a velocity due to its pressure is made to impinge upon a mass of cool water, to which it transfers its energy to such an extent that the combined jet of steam and water will overcome a resistance-head equal to or greater than the pressure which actuated the original jet of steam. That is, a jet of steam issuing from a boiler into an injector will pick up a quantity of water, and will be able to force that water into the boiler against the pressure which actuated the jet, and will carry the steam in the original jet also back into the boiler from which it started.

The injector conforming to this definition consists of a hollow, somewhat tubular casting, usually of brass, into which are made three openings. The first one (*A*, Fig. 196), which usually enters the top of the instrument, is for the delivery to it of hot, dry steam from the dome of the boiler or other convenient place. The second opening, *B*, is the inlet for the water to be fed, which is usually delivered to it from below. The third will be the feed-outlet, *HI*, opening towards the boiler, through which the feed-water impelled by the steam will pass to overcome the pressure on the check-valve and enter the boiler.

Fig. 196.

The injector properly so called has the cross-section of its tube proportions diminishing from the nozzle of the jet, in order that the velocity of the stream may increase from the point of meeting the water until the current streams into the vessel in which a high tension is maintained. If, on the other hand, the tube proportions are arranged to flare or increase in cross-section beyond the combining point, the velocity of flow is decreased, less resistance-head will be overcome, and the instrument becomes properly designated an *ejector*. The injector may therefore properly be limited to instruments operating to force water against considerable resistances and in comparatively small volumes, the water becoming considerably raised in temperature in the process. The ejector, on the other hand, is adapted for handling large volumes against low resistances, and by virtue of the greater mass of water handled the latter becomes only slightly warmed. Both injector and ejector as appliances for moving water are wasteful of heat as compared with a good pump; when applied as a boiler-feeding appliance where heating of the feed-water is convenient and desirable, the injector does
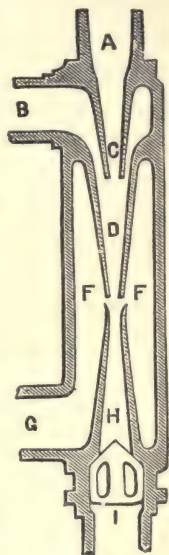
what can only be done with a pump by adding to the latter much of complication in the way of heat-saving devices.

The injector depends on three sets of principles. Two of these are physical or mechanical, attaching to it because it is a jet and impact apparatus; the third are the thermal or heat principles, resulting from the heat-transfers when steam and water are the fluids concerned.

342. Mechanical Principles underlying the Injector. The Induced-current Principle.— The injector must be capable of lifting the water which it is to feed into the boiler from a level in a tank lower than the instrument itself. If circumstances permit the water to flow to it from a higher level, it does not require to use this capacity, and will be called a non-lifting injector. Where the machine must raise the water to its level it will be called a lifting injector.

The principle on which the injector depends for its capacity to lift water is known in pneumatics or hydraulics as the principle of induced currents. If a jet of steam or air or water is made to move with a considerable velocity in a line parallel to the axis of a second or larger tube which surrounds the issuing jet, the impact of the matter issuing from the moving jet upon the matter within the surrounding tube will cause the contents of the latter to move with the jet in its direction. The action may be an impact of particles, or it may be a frictional entrainment of the one fluid by the movement of the other. If the cross-section or profile of the inner jet and the outer tube be adjusted to each other in the light of experience, the jet will induce a continued flow, tending to exhaust the contents in the space in the outer tube B which lies behind the orifice of the inducing jet C. This principle is a familiar one in the exhaust-steam blast in the locomotive, in the atomizer of the chemist and physician, and in many much-used applications. As applied in the lifting-injector it requires that the space behind the nozzle of the operative jet be connected by an air-tight pipe to the suction-

tank, within which it must be so immersed as to be water-sealed. In front of the nozzle of the jet must be an orifice opening to the air or to some waste-connection. With this latter orifice open, if the operating-jet be started, it will carry with it the air behind the jet until the pressure between the jet and the suction-tank becomes so much less than the pressure of the free atmosphere that the latter forces water up the suction-pipe to maintain the equilibrium. If the suction-pipe be not too long—probably less than 20 feet of the possible 32 feet—and the water in the suction-pipe is not so warm as to form a vapor in it under the reduced pressure, the induced current of air will cause the water to rise and meet the impact of the steam at the issue from the nozzle.

The second mechanical principle is the principle of impact of a small mass of steam in the jet against the greater mass of the water which the induced current has lifted. This will be discussed in a succeeding paragraph.

343. **Heat-transfer, Work, and Efficiency in the Injector.**—The injector problem usually comes in the form that a certain weight of water W_w is to be forced into the boiler, and to do this work a weight of steam W_s must be expended. The weight of water will have its temperature raised from T_4, which it had in the tank, to the temperature T_2, which will be the temperature of the hot feed-water after leaving the injector, or $W_w(T_2 - T_4)$. The weight of steam condensed by this weight of water loses an amount of heat per pound which is represented by the equation

$$L = C(T_1 - T_2) + xr,$$

in which the specific heat may be called unity, and the percentage of vaporized steam is 100, or x equals unity, and r is the heat of vaporization at the temperature T_1 in the boiler, at which temperature the steam enters the injector. For a weight of steam W_s this becomes

$$W_s h = W_s[(T_1 - T_2) + r].$$

The heat given by the steam must equal that absorbed by the water; or

$$W_w(T_2 - T_4) = W_s[(T_1 - T_2) + r],$$

whence

$$W_s = W_w \frac{t_2 - t_4}{t_1 - t_2 + r}$$

in ordinary temperatures, since differences alone are used. The temperature t_2 of the water in the pipe leaving the injector must be low enough to have the condensation complete, and yet the hotter it is the better so far as the boiler is concerned. It is likely to be about 160° with small or light feeding and 120° to 140° with heavier feeding. According as the upper or lower values are taken, a calculation within ordinary ranges of pressures will bring a relation of steam weight to weight of water ranging between 10 and 13. That is, the steam supplies from 10 to 13 times its weight of water. In the absence of tables for r, it may be calculated from the formula

$$r = a - bT_1 = 1114.4 - 0.7T_1.$$

The water and the steam unite and flow together through the feed-pipe. The work to be done in the boiler where the water passes the check-valve will be the displacing per unit of time of a volume in cubic feet which is that of $W_w + W_s$ pounds against the boiler-pressure. Since the pressure of the atmosphere is exerted within the injector (p_0), the effective pressure to be overcome at the boiler is $p_1 - p_0$. The volume of $W_w + W_s$ in cubic feet will be

$$\frac{W_w + W_s}{62.4} = 0.016(W_w + W_s);$$

hence the work will be, if pressures are in pounds per square inch,

Work $= 144(p_1 - p_0) \times 0.016(W_w + W_s)$ foot-pounds.

From this it is easy to pass to the efficiency,

$$E = \frac{\text{Work done}}{\text{Heat expanded}} = \frac{144(p_1 - p_0) \times 0.016(W_w + W_s)}{778\,W_s(T_1 - T_2 + xr)},$$

which as a rule works out a small value only.

344. Mechanical Principle of Impact in the Injector.— The mass of steam which meets the water in the combining tube of the injector has to act upon the latter by impact of the condensed water upon the feed-water. These masses being related to each other in a high ratio, such as one of steam to ten or sixteen of water, it follows that the principle of the conservation of the motion of the centre of gravity will bring about a resultant velocity when they meet which will be to the velocity of the steam as W_s is to W_w. That is, if the mass of the steam be one and that of the water be ten, then the centre of gravity of the two bodies will lie nearest the water and at one eleventh of the distance separating them, and the velocity of the combination after impact will be one eleventh of that of the steam. If the ratio of masses be 15, then the resultant velocity will be $\frac{1}{16}$ of that of the steam.

The accepted formulæ for the velocity of flow of a perfect gas from a reservoir within which is a pressure p_1 into another chamber where the pressure p_2 prevails are those of Joule and Thomson. When the weight of a unit of volume is denoted by w_1 at the pressure p_1 and the cooling during discharge is adiabatic, then

$$V^2 = 2g\frac{k}{(k-1)}\frac{p_1}{w_1}\left[1 - \frac{T_2}{T_1}\right] = 2g\frac{k}{k-1}p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right],$$

in which k will have the value appropriate for the observed adiabatic expansion of the gas in question. For steam it is 1.111 according to Rankine, and according to Zeuner the value

$$k = 1.035 + 0.100x,$$

in which x is the initial proportion of dry steam-gas. Solving, with the assumption that the steam is dry,

$$V = 23.2687 \sqrt{p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{0.1189} \right]}.$$

The velocity of the mixture of steam and water will be less than the one-tenth or one-sixteenth of the foregoing values, because the impact of all particles cannot be in the line of propulsion. Calling this water velocity V_2, the cross-section of the water-tube inside the injector will be

$$\text{Tube area} = \frac{0.016(W_w + W_s)}{V_2}.$$

The area of the steam-nozzle should be

$$\text{Nozzle area} = \frac{\text{Volume of steam corresponding to } W_s \text{ per sec.}}{V},$$

from which the diameters will follow since the area will be πr^2.

It will be found, on making the calculation for any case, that the expenditure of heat-units resulting from the condensation process is so much greater than the equivalent expenditure in lifting the water by suction and of forcing it into the boiler, that these latter quantities are negligible by comparison.

345. Double-tube Injector. The Inspirator. —The necessity of adjusting the weight of water to the weight of steam in the jet, and the variation in the latter with varying steam-pressure, brought about the self-adjusting types of injector, where the combining tube was moved forward or backward, enlarging or constricting the water-area as the pressure outside of the tube might vary. This was a feature of improvements by Sellers & Co. as far back as 1865 (Fig. 197). In 1876–7 Körting of Hanover, Germany, (Fig. 198,) and Hancock of Boston introduced the use of two jets of steam to

secure this same result. The first nozzle is the smaller and acts upon a relatively large mass of water to deliver it under a slight pressure to the second or larger steam-nozzle, which forces the supply to the boiler. The first jet will be the
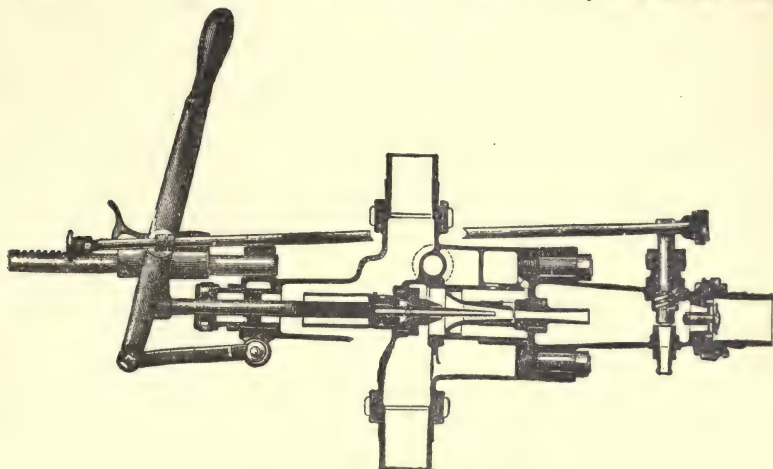
FIG. 197.
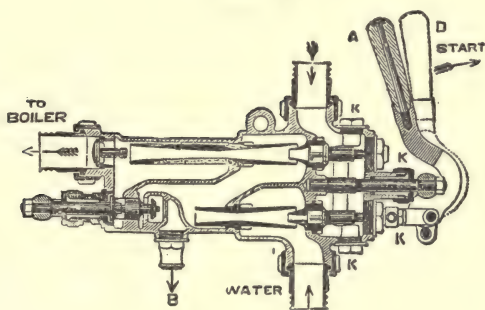
FIG. 198.

lifting-jet when the injector requires to raise its water. Inspirator was the proprietary name given by Hancock to his double-tube design.

346. Restarting or Automatic Injectors.—An injector which will establish its action as a boiler-feeder automatically after the continuity of the combined jet has been broken by

a stoppage of either the steam- or water-supply is called an automatic or restarting injector. The usual method of securing it is to have two steam-jets, one for lifting and one for forcing. When the continuity of the combined jet is broken by a failure of the water-supply, the discharge of steam from the forcing-jet finds its way to a waste-pipe through a check-valve, while the lifting-jet keeps up maintaining a vacuum in the feed-pipe and draws up the water as soon as it can be supplied. The adjustment of tubes and nozzles is also so made as to favor a wide variation of steam-pressures before a break in the flow shall occur. The advantage of the restarting principle is the simplification of the apparatus by doing away with adjusting spindles and the like, so that a simple valve on the steam-pipe to the injector is all the regulating appliance required.

347. Exhaust-steam Injectors.—The steam-current for an injector may be derived from exhaust-steam when the back-pressure head is less than 75 pounds. The feed-water should flow to the apparatus and should be as cool as possible —never over 100° F. If a supplemental live-steam jet be added to revivify the exhaust-jet, the injector will feed up to 150 pounds.

348. Advantages of the Injector.—The advantages to be claimed for the injector are:

1. From its construction, it is cheap.

2. It is compact, and takes little space in proportion to its capacity for moving water.

3. It has few or no moving parts, and hence a small running cost for repairs.

4. It delivers the water warm to the boiler.

5. It has no exhaust-steam to dispose of.

349. Disadvantages of the Injector.—The disadvantages of the injector are those which belong to the apparatus as a class, and those which belong to certain forms of the instrument only.

1. The impact of the jet on the water is not an effective method of pumping. As a pumping appliance the injector is about $\frac{1}{5}$ as efficient as the equivalent steam-pump. Its duty is about 2 million foot-pounds, against 10 million for the pump.

2. It heats its water with live steam, while to utilize the heat of its exhaust a pump with feed-water heater should show a superior coal economy of 12 per cent from the saving of waste heat.

3. The feed-water must be cool enough to condense the steam-jet, and this limit is about 100° F. Hence the injector cannot be used upon hot water.

4. It will not start with pressures much lower than those for which it was designed.

5. If it is not a restarting instrument, it will stop working after the limit of co-relation of feed-water and steam are passed. Then it must be started anew by the operator. Often when it has become hot by the interruption of the water-jet, it can only be restarted by complete cooling with water.

APPENDIX.

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LOGARITHMS.

35c. In arithmetical computations, the usual base of the system is 10, so that x, the logarithm for a number m, will be the exponent to which 10 is to be raised to give the quantity m, or $x = \log_{10} m$. In analytical mathematical work, the base generally employed is not 10, but is represented by e whose value is 2.71828 +. To convert common or Briggs logarithms into Napierian logarithms, the former are to be multiplied by 2.3026.

The equation of the hyperbola in the form $xy = $ constant leads to the deduction that the area between the hyperbolic curve and its nearest asymptote cut off by two ordinates parallel to the other asymptote and distant respectively from the origin by a and b will be proportional to $\log \dfrac{b}{a}$. Hence it will be true that the integral of $\dfrac{dx}{x}$ will be the hyperbolic logarithm of x. To save trouble of conversion, a table is appended covering the usual ranges required.

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.01	.0099	1.20	.1823	1.39	.3293	1.58	.4574	1.77	.5710
1.02	.0198	1.21	.1906	1.40	.3365	1.59	.4637	1.78	.5766
1.03	.0296	1.22	.1988	1.41	.3436	1.60	.4700	1.79	.5822
1 04	.0392	1·23	.2070	1.42	.3507	1.61	.4762	1.80	.5878
1.05	.0488	1.24	.2151	1.43	.3577	1.62	.4824	1.81	.5933
1.06	.0583	1.25	.2231	1.44	.3646	1 63	.4886	1.82	.5988
1.07	.0677	1.26	.2311	1.45	.3716	1.64	.4947	1 83	.6043
1.08	.0770	1.27	.2390	1.46	.3784	1.65	.5008	1 84	.6098
1.09	.0862	1.28	.2469	1.47	.3853	1.66	.5068	1.85	.6152
1.10	.0953	1.29	.2546	1.48	.3920	1.67	.5128	1.86	.6206
1.11	.1044	1.30	.2624	1.49	.3988	1.68	.5188	1.87	.6259
1.12	.1133	1.31	.2700	1.50	.4055	1.69	.5247	1.88	.6313
1.13	.1222	1.32	.2776	1.51	.4121	1.70	.5306	1.89	.6366
1 14	.1310	1.33	.2852	1.52	.4187	1.71	.5365	1.90	.6419
1.15	.1398	1.34	.2927	1.53	.4253	1.72	.5423	1.91	.6471
1.16	.1484	1.35	.3001	1.54	.4318	1.73	.5481	1 92	.6523
1.17	.1570	1.36	.3075	1.55	.4383	1.74	.5539	1.93	.6575
1.18	.1655	1.37	.3148	1.56	.4447	1.75	.5596	1.94	.6627
1.19	.1740	1.38	.3221	1.57	.4511	1.76	.5653	1.95	.6678

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.96	.6729	2.66	.9783	3.36	1.2119	4.06	1.4012	4.76	1.5602
1.97	.6780	2.67	.9821	3.37	1.2149	4.07	1.4036	4.77	1.5623
1.98	.6831	2.68	.9858	3.38	1.2179	4.08	1.4061	4.78	1.5644
1.99	.6881	2.69	.9895	3.39	1.2208	4.09	1.4085	4.79	1.5665
2.00	.6931	2.70	.9933	3.40	1.2238	4.10	1.4110	4.80	1.5686
2.01	.6981	2.71	.9969	3.41	1.2267	4.11	1.4134	4.81	1.5707
2.02	.7031	2.72	1.0006	3.42	1.2296	4.12	1.4159	4.82	1.5728
2.03	.7080	2.73	1.0043	3.43	1.2326	4.13	1.4183	4.83	1.5748
2.04	.7129	2.74	1.0080	3.44	1.2355	4.14	1.4207	4.84	1.5769
2.05	.7178	2.75	1.0116	3.45	1.2384	4.15	1.4231	4.85	1.5790
2.06	.7227	2.76	1.0152	3.46	1.2413	4.16	1.4255	4.86	1.5810
2.07	.7275	2.77	1.0188	3.47	1.2442	4.17	1.4279	4.87	1.5831
2.08	.7324	2.78	1.0225	3.48	1.2470	4.18	1.4303	4.88	1.5851
2.09	.7372	2.79	1.0260	3.49	1.2499	4.19	1.4327	4.89	1.5872
2.10	.7419	2.80	1.0296	3.50	1.2528	4.20	1.4351	4.90	1.5892
2.11	.7467	2.81	1.0332	3.51	1.2556	4.21	1.4375	4.91	1.5913
2.12	.7514	2.82	1.0367	3.52	1.2585	4.22	1.4398	4.92	1.5933
2.13	.7561	2.83	1.0403	3.53	1.2613	4.23	1.4422	4.93	1.5953
2.14	.7608	2.84	1.0438	3.54	1.2641	4.24	1.4446	4.94	1.5974
2.15	.7655	2.85	1.0473	3.55	1.2669	4.25	1.4469	4.95	1.5994
2.16	.7701	2.86	1.0508	3.56	1.2698	4.26	1.4493	4.96	1.6014
2.17	.7747	2.87	1.0543	3.57	1.2726	4.27	1.4516	4.97	1.6034
2.18	.7793	2.88	1.0578	3.58	1.2754	4.28	1.4540	4.98	1.6054
2.19	.7839	2.89	1.0613	3.59	1.2782	4.29	1.4563	4.99	1.6074
2.20	.7885	2.90	1.0647	3.60	1.2809	4.30	1.4586	5.00	1.6094
2.21	.7930	2.91	1.0682	3.61	1.2837	4.31	1.4609	5.01	1.6114
2.22	.7975	2.92	1.0716	3.62	1.2865	4.32	1.4633	5.02	1.6134
2.23	.8020	2.93	1.0750	3.63	1.2892	4.33	1.4656	5.03	1.6154
2.24	.8065	2.94	1.0784	3.64	1.2920	4.34	1.4679	5.04	1.6174
2.25	.8109	2.95	1.0513	3.65	1.2947	4.35	1.4702	5.05	1.6194
2.26	.8154	2.96	1.0852	3.66	1.2975	4.36	1.4725	5.00	1.6214
2.27	.8198	2.97	1.0886	3.67	1.3002	4.37	1.4748	5.07	1.6233
2.28	.8242	2.98	1.0919	3.68	1.3029	4.38	1.4770	5.08	1.6253
2.29	.8286	2.99	1.0953	3.69	1.3056	4.39	1.4793	5.09	1.6273
2.30	.8329	3.00	1.0986	3.70	1.3083	4.40	1.4816	5.10	1.6292
2.31	.8372	3.01	1.1019	3.71	1.3110	4.41	1.4839	5.11	1.6312
2.32	.8416	3.02	1.1053	3.72	1.3137	4.42	1.4861	5.12	1.6332
2.33	.8458	3.03	1.1086	3.73	1.3164	4.43	1.4884	5.13	1.6351
2.34	.8502	3.04	1.1119	3.74	1.3191	4.44	1.4907	5.14	1.6371
2.35	.8544	3.05	1.1151	3.75	1.3218	4.45	1.4929	5.15	1.6390
2.36	.8587	3.06	1.1184	3.76	1.3244	4.46	1.4951	5.16	1.6409
2.37	.8629	3.07	1.1217	3.77	1.3271	4.47	1.4974	5.17	1.6429
2.38	.8671	3.08	1.1249	3.78	1.3297	4.48	1.4996	5.18	1.6448
2.39	.8713	3.09	1.1282	3.79	1.3324	4.49	1.5019	5.19	1.6467
2.40	.8755	3.10	1.1314	3.80	1.3350	4.50	1.5041	5.20	1.6487
2.41	.8796	3.11	1.1346	3.81	1.3376	4.51	1.5063	5.21	1.6506
2.42	.8838	3.12	1.1378	3.82	1.3403	4.52	1.5085	5.22	1.6525
2.43	.8879	3.13	1.1410	3.83	1.3429	4.53	1.5107	5.23	1.6544
2.44	.8920	3.14	1.1442	3.84	1.3455	4.54	1.5129	5.24	1.6563
2.45	.8961	3.15	1.1474	3.85	1.3481	4.55	1.5151	5.25	1.6582
2.46	.9002	3.16	1.1506	3.86	1.3507	4.56	1.5173	5.26	1.6601
2.47	.9042	3.17	1.1537	3.87	1.3533	4.57	1.5195	5.27	1.6620
2.48	.9083	3.18	1.1569	3.88	1.3558	4.58	1.5217	5.28	1.6639
2.49	.9123	3.19	1.1600	3.89	1.3584	4.59	1.5239	5.29	1.6658
2.50	.9163	3.20	1.1632	3.90	1.3610	4.60	1.5261	5.30	1.6677
2.51	.9203	3.21	1.1663	3.91	1.3635	4.61	1.5282	5.31	1.6696
2.52	.9243	3.22	1.1694	3.92	1.3661	4.62	1.5304	5.32	1.6715
2.53	.9282	3.23	1.1725	3.93	1.3686	4.63	1.5326	5.33	1.6734
2.54	.9322	3.24	1.1756	3.94	1.3712	4.64	1.5347	5.34	1.6752
2.55	.9361	3.25	1.1787	3.95	1.3737	4.65	1.5369	5.35	1.6771
2.56	.9400	3.26	1.1817	3.96	1.3762	4.66	1.5390	5.36	1.6790
2.57	.9439	3.27	1.1848	3.97	1.3788	4.67	1.5412	5.37	1.6808
2.58	.9478	3.28	1.1878	3.98	1.3813	4.68	1.5433	5.38	1.6827
2.59	.9517	3.29	1.1909	3.99	1.3838	4.69	1.5454	5.39	1.6845
2.60	.9555	3.30	1.1939	4.00	1.3863	4.70	1.5476	5.40	1.6864
2.61	.9594	3.31	1.1969	4.01	1.3888	4.71	1.5497	5.41	1.6882
2.62	.9632	3.32	1.1999	4.02	1.3913	4.72	1.5518	5.42	1.6901
2.63	.9670	3.33	1.2030	4.03	1.3938	4.73	1.5539	5.43	1.6919
2.64	.9708	3.34	1.2060	4.04	1.3962	4.74	1.5560	5.44	1.6938
2.65	.9746	3.35	1.2090	4.05	1.3987	4.75	1.5581	5.45	1.6956

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
5.46	1.6974	6.16	1.8181	6.86	1.9257	7.56	2.0229	8.32	2.1187
5.47	1.6993	6.17	1.8197	6.87	1.9272	7.57	2.0242	8.34	2.1211
5.48	1.7011	6.18	1.8213	6.88	1.9286	7.58	2.0255	8.36	2.1235
5.49	1.7029	6.19	1.8229	6.89	1.9301	7.59	2.0268	8.38	2.1258
5.50	1.7047	6.20	1.8245	6.90	1.9315	7.60	2.0281	8.40	2.1282
5.51	1.7066	6.21	1.8262	6.91	1.9330	7.61	2.0295	8.42	2.1306
5.52	1.7084	6.22	1.8278	6.92	1.9344	7.62	2.0308	8.44	2.1330
5.53	1.7102	6.23	1.8294	6.93	1.9359	7.63	2.0321	8.46	2.1353
5.54	1.7120	6.24	1.8310	6.94	1.9373	7.64	2.0334	8.48	2.1377
5.55	1.7138	6.25	1.8326	6.95	1.9387	7.65	2.0347	8.50	2.1401
5.56	1.7156	6.26	1.8342	6.96	1.9402	7.66	2.0360	8.52	2.1424
5.57	1.7174	6.27	1.8358	6.97	1.9416	7.67	2.0373	8.54	2.1448
5.58	1.7192	6.28	1.8374	6.98	1.9430	7.68	2.0386	8.56	2.1471
5.59	1.7210	6.29	1.8390	6.99	1.9445	7.69	2.0399	8.58	2.1494
5.60	1.7228	6.30	1.8405	7.00	1.9459	7.70	2.0412	8.60	2.1518
5.61	1.7246	6.31	1.8421	7.01	1.9473	7.71	2.0425	8.62	2.1541
5.62	1.7263	6.32	1.8437	7.02	1.9488	7.72	2.0438	8.64	2.1564
5.63	1.7281	6.33	1.8453	7.03	1.9502	7.73	2.0451	8.66	2.1587
5.64	1.7299	6.34	1.8469	7.04	1.9516	7.74	2.0464	8.68	2.1610
5.65	1.7317	6.35	1.8485	7.05	1.9530	7.75	2.0477	8.70	2.1633
5.66	1.7334	6.36	1.8500	7.06	1.9544	7.76	2.0490	8.72	2.1656
5.67	1.7352	6.37	1.8516	7.07	1.9559	7.77	2.0503	8.74	2.1679
5.68	1.7370	6.38	1.8532	7.08	1.9573	7.78	2.0516	8.76	2.1702
5.69	1.7387	6.39	1.8547	7.09	1.9587	7.79	2.0528	8.78	2.1725
5.70	1.7405	6.40	1.8563	7.10	1.9601	7.80	2.0541	8.80	2.1748
5.71	1.7422	6.41	1.8579	7.11	1.9615	7.81	2.0554	8.82	2.1770
5.72	1.7440	6.42	1.8594	7.12	1.9629	7.82	2.0567	8.84	2.1793
5.73	1.7457	6.43	1.8610	7.13	1.9643	7.83	2.0580	8.86	2.1815
5.74	1.7475	6.44	1.8625	7.14	1.9657	7.84	2.0592	8.88	2.1838
5.75	1.7492	6.45	1.8641	7.15	1.9671	7.85	2.0605	8.90	2.1861
5.76	1.7509	6.46	1.8656	7.16	1.9585	7.86	2.0618	8.92	2.1883
5.77	1.7527	6.47	1.8672	7.17	1.9699	7.87	2.0631	8.94	2.1905
5.78	1.7544	6.48	1.8687	7.18	1.9713	7.88	2.0643	8.96	2.1928
5.79	1.7561	6.49	1.8703	7.19	1.9727	7.89	2.0656	8.98	2.1950
5.80	1.7579	6.50	1.8718	7.20	1.9741	7.90	2.0669	9.00	2.1972
5.81	1.7596	6.51	1.8733	7.21	1.9754	7.91	2.0681	9.02	2.1994
5.82	1.7613	6.52	1.8749	7.22	1.9769	7.92	2.0694	9.04	2.2017
5.83	1.7630	6.53	1.8764	7.23	1.9782	7.93	2.0707	9.06	2.2039
5.84	1.7647	6.54	1.8779	7.24	1.9796	7.94	2.0719	9.08	2.2061
5.85	1.7664	6.55	1.8795	7.25	1.9810	7.95	2.0732	9.10	2.2083
5.86	1.7681	6.56	1.8810	7.26	1.9824	7.96	2.0744	9.12	2.2105
5.87	1.7699	6.57	1.8825	7.27	1.9838	7.97	2.0757	9.14	2.2127
5.88	1.7716	6.58	1.8840	7.28	1.9851	7.98	2.0769	9.16	2.2148
5.89	1.7733	6.59	1.8856	7.29	1.9865	7.99	2.0782	9.18	2.2170
5.90	1.7750	6.60	1.8871	7.30	1.9879	8.00	2.0794	9.20	2.2192
5.91	1.7766	6.61	1.8886	7.31	1.9892	8.01	2.0807	9.22	2.2214
5.92	1.7783	6.62	1.8901	7.32	1.9906	8.02	2.0819	9.24	2.2235
5.93	1.7800	6.63	1.8916	7.33	1.9920	8.03	2.0832	9.26	2.2257
5.94	1.7817	6.64	1.8931	7.34	1.9933	8.04	2.0844	9.28	2.2279
5.95	1.7834	6.65	1.8946	7.35	1.9947	8.05	2.0857	9.30	2.2300
5.96	1.7851	6.66	1.8961	7.36	1.9961	8.06	2.0869	9.32	2.2322
5.97	1.7867	6.67	1.8976	7.37	1.9974	8.07	2.0882	9.34	2.2343
5.98	1.7884	6.68	1.8991	7.38	1.9988	8.08	2.0894	9.36	2.2364
5.99	1.7901	6.69	1.9006	7.39	2.0001	8.09	2.0906	9.38	2.2386
6.00	1.7918	6.70	1.9021	7.40	2.0015	8.10	2.0919	9.40	2.2407
6.01	1.7934	6.71	1.9036	7.41	2.0028	8.11	2.0931	9.42	2.2428
6.02	1.7951	6.72	1.9051	7.42	2.0041	8.12	2.0943	9.44	2.2450
6.03	1.7967	6.73	1.9066	7.43	2.0055	8.13	2.0956	9.46	2.2471
6.04	1.7984	6.74	1.9081	7.44	2.0069	8.14	2.0968	9.48	2.2492
6.05	1.8001	6.75	1.9095	7.45	2.0082	8.15	2.0980	9.50	2.2513
6.06	1.8017	6.76	1.9110	7.46	2.0096	8.16	2.0992	9.52	2.2534
6.07	1.8034	6.77	1.9125	7.47	2.0108	8.17	2.1005	9.54	2.2555
6.08	1.8050	6.78	1.9140	7.48	2.0122	8.18	2.1017	9.56	2.2576
6.09	1.8066	6.79	1.9155	7.49	2.0136	8.19	2.1029	9.58	2.2597
6.10	1.8083	6.80	1.9169	7.50	2.0149	8.20	2.1041	9.60	2.2618
6.11	1.8099	6.81	1.9184	7.51	2.0162	8.22	2.1066	9.62	2.2638
6.12	1.8116	6.82	1.9199	7.52	2.0176	8.24	2.1090	9.64	2.2659
6.13	1.8132	6.83	1.9213	7.53	2.0189	8.26	2.1114	9.66	2.2680
6.14	1.8148	6.84	1.9228	7.54	2.0202	8.28	2.1138	9.68	2.2701
6.15	1.8165	6.85	1.9242	7.55	2.0215	8.30	2.1163	9.70	2.2721

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
9.72	2.2742	10.25	2.3279	14.00	2.6391	21	3.0445	36	3.5835
9.74	2.2762	10.50	2.3513	14.25	2.6567	22	3.0910	37	3.6109
9.76	2.2783	10.75	2.3749	14.50	2.6740	23	3.1355	38	3.6376
9.78	2.2803	11.00	2.3979	14.75	2.6913	24	3.1781	39	3.6636
9.80	2.2824	11.25	2.4201	15.00	2.7081	25	3.2189	40	3.6889
9.82	2.2844	11.50	2.4430	15.50	2.7408	26	3.2581	41	3.7136
9.84	2.2865	11.75	2.4636	16.00	2.7726	27	3.2958	42	3.7377
9.86	2.2885	12.00	2.4849	16.50	2.8034	28	3.3322	43	3 7612
9.88	2.2905	12.25	2.5052	17.00	2.8332	29	3.3673	44	3.7842
9.90	2.2925	12.50	2.5262	17.50	2.8621	30	3.4012	45	3.8067
9.92	2.2946	12.75	2.5455	18.00	2.8904	31	3.4340	46	3.8286
9.94	2.2966	13.00	2.5649	18.50	2.9173	32	3.4657	47	3.8501
9.96	2.2986	13.25	2.5840	19.00	2.9444	33	3.4965	48	3.8712
9.98	2.3006	13.50	2.6027	19.50	2.9703	34	3.5263	49	3.8918
10.00	2.3026	13.75	2.6211	20.00	2.9957	35	3.5553	50	3.9120

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house ; the Civil Engineers of Great Britain, and the American Society
of Mechanical Engineers.

353. THE NAMES OF DISTINGUISHED SCIENTISTS and investigators (mainly
physicists), who have contributed to the present knowledge of heat, group
themselves into two classes.   The first contains those who have made
a specialty of heat, and the second includes those who have studied it in
connection with other lines of research.   The lists are by no means ex-
haustive, and include of course some of the writers in the preceding lists.

GROUP I.

Balfour-Stewart, Berthelot, Boyle, Charles, Colding, Davy, Dufour,
Dulong, Favre, Forbes, Franz, Gay-Lussac, Helmholtz, Hopkins, Joule,
Le Chatelier, Leslie, Mahler, Mariotte, Mayer, Melloni, Peclet, Petit,
Plank, Playfair, Pouillet, Regnault, Rumford, Silbermann, Tait, Tyndall,
Weidemann.

GROUP II.

Andrews, Angström, Arago, Béclard, Bérard, Bérnouilli, Bianchi,
Black, Bunsen, Cauchy, Clapeyron, Dalton, De Candolle, de la Provostaye,
de la Rive, de la Roche, Deluc, Desains, Despretz, Deville, Donny, Dumas,
Fairbairn, Faraday, Fizeau, Foucault, Gandin, Grove, Guthrie, Hallström,
Hallauer, Hess, Hirn, Kirchoff, Knoblauch, Krönig, Kupffer, Laplace,
Lavoisier, Magnus, Mallard, Marcet, Masson, Moll, Mousson, Morin,
Neumann, Person, Pictet, Poisson, Prevost, Redtenbacher, Rudberg,
Saussure, Séguin, Tate, Thompson, Thomson, Tresca, Troost, Van Beck,
Woestyn.

354. NOTES AND REFERENCES.

10. See Heat as a Mode of Motion. John Tyndall, 1864, p. 85, also Joule,
Mayer and Rowland in Bibliography.

11. Consult Trans. A. S. M. E., vol. 13, pp. 351–358.

12. Specific Heats.   Consult Ganot's Physics, p. 341, also Röntgen-
Dubois, *Thermodynamics*, pp. 134–137. Regnault's determinations. Average
values, taken between freezing and boiling points.   Water being unity.

SOLIDS.

Antimony	0.05077	Cadmium	0.05670
Copper	0.09515	Steel (soft)	0.11650
Gold	0.03244	Steel (hard)	0.11750
Wrought iron	0.11379	Zinc	0.09555
Glass	0.19768	Brass	0.09391
Cast iron (gray)	0.12983	Ice	0.50400
Lead	0.03140	Sulphur	0.20259
Platinum	0.03244	Charcoal	0.24111
Silver	0.05701	Alumina	0.19700
Tin	0.05623	Phosphorus	0.18949
Cobalt	0.10696	Nickel	0.10863
Mercury	0.03332	Phosphite	0.020107
Bismuth	0.03084	Arsenic	0.08140

From other sources (Kent).

Brick-work and masonry (about)	0.200	Magnesia	0.222
Marble	0.210	Soda	0.231
Chalk	0.215	Corundum	0.198
Quicklime	0.217	Gypsum	0.197
Silica	0.191	Stones generally	0.220
Pine wood	0.467	Oak wood	0.570
Fir wood	0.650	Pear wood	0.500

LIQUIDS.

Water	1.0000	Mercury	0.0333
Melted lead	0.0402	Alcohol	0.7000
" sulphur	0.2340	Fusel oil (mixed alcohols)	0.5640
" bismuth	0.0308	Benzine	0.4500
" tin	0.0637	Ether	0.5034
Sulphuric acid	0.3350	Olive oil	0.3100
Hydrochloric acid	0.6000	Turpentine	0.4720
Bromine	1.1110		

GASES. (See § 107.)

	Pressure Constant.	Volume Constant.
Air	0.23751	0.16847
Oxygen	0.21751	0.15507
Hydrogen	3.40900	2.41226
Nitrogen	0.24380	0.17273
Steam superheated	0.4805	0.346
Carbonic acid	0.217	0.1535
Olefiant gas (CH_2)	0.404	0.173

	Pressure Constant.	Volume Constant.
Carbonic oxide..........	0.2479	0.1758
Ammonia.................	0.508	0.299
Alcohol.................	0 4534	0.3200
Acetic acid.............	0.4125	......
Chloroform.............	0.1567	......
Ether..................	0.4797	0.3411
Marsh gas (CH_4).......	0.5929	0.4683
Bisulphide of carbon....	0.1570	......
Sulphuric acid.........	0.1553	......

Specific heats are greater if higher superior limit is taken than boiling-point. Specific heat of water is not constant.

$$\text{At } \quad 0° \text{ C.} = 1.0000$$
$$\text{`` } \quad 50° \text{ `` } = 1.0042$$
$$\text{`` } \quad 100° \text{ `` } = 1.0132$$
$$\text{`` } \quad 150° \text{ `` } = 1.0262$$
$$\text{`` } \quad 200° \text{ `` } = 1.0440$$
$$\text{`` } \quad 230° \text{ `` } = 1.0568$$

14. To convert readings in F. degrees to equivalents in C. degrees

$$\text{C.} = (F - 32)\tfrac{5}{9}$$
$$\text{F.} = \tfrac{9}{5}C + 32$$

Care must be taken to observe signs when converting below the freezing-point.

Réaumur's conversions use $\tfrac{4}{5}$ and $\tfrac{9}{4}$ respectively. Little used.

16. Joule and Thomson's investigations. Phil. Trans. vol. 144, p. 349, 1854. Rowland's investigations. Proc. Am. Acad., vol. 15 (N. S. 8), p. 75, 1879.

ABSOLUTE AND AIR THERMOMETER SCALES COMPARED.   CENTIGRADE.

Air Thermometer Reads.	Absolute Scale should be	Correct Air Scale Reading by
0	0	0
10	9.9972	— 0.0028
20	19.9952	— 0.0048
30	29.9939	— 0.0061
40	39.9933	— 0.0067
50	49.9932	— 0.0068
60	59.9937	— 0.0063
70	69.9946	— 0.0054
80	79.9956	— 0.0042
90	89.9978	— 0.0022
100	100.000	+ 0.
200	200.037	+ 0.037
300	300.092	+ 0.092
400	400.157	+ 0.157
500	500.228	+ 0.228

23. See investigations by **Scheurer-Kestner** and Meunier-Dollfus, Bulletin Soc. Indust. de Mulhouse (1868). Also Mahler's Test of Coals, *Mineral Industry*, vol. I, p. 97.

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39. For tests of calorimetric power of bagasse see paper by Dr. W. O. Atwater, quoted in Sturtevant *Mechanical Draft*, pp. 39–42. For furnaces, see *ibid.*, pp. 310, 366, 371. Also L. A. Bocuel, Louisiana Sugar Chemists Assoc. 1892.

48–49. Hutton's Mechanical Engineering of Power Plants, pp. 567–573. Also Trans. Inst. Mech. Eng. of Gt. Btn. 1889. See also E. A. Brayley-Hodgetts, Liquid Fuel.

(*Iron Age*, Nov. 2, 1893.)—Test by the Twin City Rapid Transit Company of Minneapolis and St. Paul. This test showed that with the ordinary Lima oil weighing $6\frac{6}{10}$ pounds per gallon, and costing $2\frac{1}{4}$ cents per gallon, and coal that gave an evaporation of $7\frac{1}{2}$ lbs. of water per pound of coal, the two fuels were equally economical when the price of coal was $3.85 per ton of 2000 lbs. With the same coal at $2.00 per ton, the coal was 37% more economical, and with the coal at $4.85 per ton, the coal was 20% more expensive than the oil. These results include the difference in the cost of handling the coal, ashes, and oil.

In 1892 there were reported to the Engineers' Club of Philadelphia some comparative figures, from tests undertaken to ascertain the relative value of coal, petroleum, and gas :

	Lbs. Water, from and at 212° F.
1 lb. anthracite coal evaporated	9.70
1 lb. bituminous coal	10.14
1 lb. free oil, 36° gravity	16.48
1 cubic foot gas, 20 C. P.	1.28

The gas used was that obtained in the distillation of petroleum, having about the same fuel-value as natural or coal gas of equal candle-power.

Taking the efficiency of bituminous coal as a basis, the calorific energy of petroleum is more than 60% greater than that of coal ; whereas, theoretically, petroleum exceeds coal only about 45%—the one containing 14,500 heat-units and the other 21,000.

Crude Petroleum vs. Indiana Block Coal for Steam-raising at the South Chicago Steel Works. (E. C. Potter, Trans. A. I. M. E., xvii., 807.)—With coal, 14 tubular boilers 16 ft. × 5 ft. required 25 men to operate them ; with fuel-oil, 6 men were required, a saving of 19 men at $2 per day, or $38 per day.

For one week's work **2731** barrels of oil were used, against 848 tons of coal required for the same work, showing 322 barrels of oil to be equivalent to 1 ton of coal. With oil at 60 cents per barrel and coal at $2.15 per ton.

the relative cost of oil to coal is as $1.93 to $2.15. No evaporation tests were made.

50. *Ibid.* See also *Helios*, E. D. Meier, St. Louis, 1895. Also Colliery Engineer, 1889–90, articles by F. T. Rowan. Also D. K. Clark, Treatise on Steam-engine.

52. Producer Gas. See Trans. Am. Inst. Mining Eng., vol. 18, p. 205. W. J. Taylor. Also Kent's Pocket-book, p. 646.

58. See Hoadley, Trials of a Warm-blast Apparatus. Trans. A. S. M. E., vol. 6, p. 676.

59. Helios, E. D. Meier, St. Louis, Mo., 1895.

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62–68. Consult H. M. Howe. Metallurgy of Steel.

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73–74. Consult Péclet, *Traité de la Chaleur*, 3ᵉ ed. 1860, p. 217 ; also W. M. Barr ; W. W. Christie.

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Webb,	"	"	"	11, p. 762.
"	"	"	"	11, p. 772.
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Thurston,	"	"	"	12, p. 85.
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78. Peabody-Miller, Notes on Steam Boilers, Boston, 1894.

79. Steam. Babcock and Wilcox Co., New York, 1893.

83. *Closed Stokeholes*. Richard Sennett. Inst. Naval Archts., London, 1886. *Steamships and their Machinery*. J. W. C. Haldane, London and New York, 1893.

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86. Mechanical Draft, W. R. Roney. Trans. A. S. M. E., vol. 15. See also Mechanical Engineering of Power Plants, Hutton, pp. 527–535.

88. See Whitham, *Effect of Retarders*. Trans. A. S. M. E., vol. 17, p. 450, No. 687.

89. See Reports on Smoke Prevention, Journal of Assoc. Engineering Societies, vol. 11, June, 1892, p. 201. See also *Iron Age*, April 7, 1892. Also O. H. Landreth, Report to State Board of Health, of Tennessee. See also *Eng'g News*, June 8, 1893 : also Report, March 10, 1888, by C. E. Jones and C. F. White to O. N. Nelson, City Council of Chicago. See also D. R. Clark, Treatise on the Steam-engine ; see Bryan, Down-draft Furnace, Trans. A. S. M. E., vol. 16, p. 773.

96. Hutton, Mech. Eng'g of Power Plants, pp. 504 to 636.

102. Hutton, Mech. Eng'g of Power Plants, pp. 680–682. Whitham, Surface Condensers, Trans. A. S. M. E., vol. 9, p. 425. Joule, Jour. Franklin Inst., 1862, p. 36.

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124. The entropy equation for a medium such as all perfect gases can be reached by an interesting and easy process. If the exponent n be the ratio between the specific heats or $n = \dfrac{C_p}{C_v}$ we may write,

$$p_1 v_1{}^n = p_2 v_2{}^n,$$

$$(p_1 v_1)^{\frac{C_p}{C_v}} = (p_2 v_2)^{\frac{C_p}{C_v}},$$

$$\left(\frac{v_2}{v_1}\right)^{C_p} = \left(\frac{p_1}{p_2}\right)^{C_v},$$

$$C_p \text{ hyp. log } \frac{v_2}{v_1} = C_v \text{ hyp. log } \frac{p_1}{p_2}.$$

But the sum of these is the entropy, and for adiabatic expansion the entropy is constant or their difference is zero; as the equation shows.

Confusion is likely to follow an attempt to view the entropy-factor as a property of a heat medium. It is so only in the sense that π is a property of the area of the circle, when written to πr^2.

156. For high-speed *vs.* low-speed, see Mech. Eng'g of Power Plants, chap. 3.

195–205. For references on temperature-entropy, consult :

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195–205. Geo. Richmond: Trans. Am. Soc. Mech. Engrs., vol. 14, p. 183, and vol. 19, p. 477; also Eng. Soc. of Columbia Univ., 1896–97.

180. For Compressed Air and Compressors, consult:

Pernolet, Peabody, Wood, Drinker on Tunneling, Röntgen, Wm. L. Saunders, Production of Compressed Air.

224. See Thurston, Trans. A. S. M. E., vol. 18, p. 160, No. 710. Promise and Potency of High-pressure Steam, from which the diagrams are reproduced by permission.

225. For Steam-jacket, see Trials of Steam Machinery U. S. Revenue Steamers Rush, Dexter, and Dallas, U. S. Treasury Dept., Nov. 1874, by C. H. Loring and C. E. Emery; also Tests of U. S. S. Gallatin, Dec. 1874 and Jan. 1875; also paper on Cylinder Condensation and Superheating, Soc. of Arts, Mass. Inst. Tech., April 1875, by Geo. B. Dixwell; also Thurston, Trans. A. S. M. E., vol. 15, p. 779, No. 590, and vol. 17, p. 488, No. 689.

228. Non-conducting cylinder linings, Novelty Iron Works experiments, 1864–68, C. E. Emery, Trans. A. S. M. E., vol. 7, p. 375, No. 204. Also vol. 1, p. 185.

229. For Superheating, see Isherwood, Experimental Researches, vol. 2; experiments on U. S. S. Mackinaw, Eutaw, and Georgeanna; also Thurston, Trans. A. S. M. E. Also M. G. Leloutre, Comptes Rendus de l'Académie de France, Dec. 21, 1891, and Soc. des Ingenieurs Civils de France, Jan. 8, 1892.

242. For Hirn's Analysis, see Hirn, Théorie Mécanique de la Chaleur, 1876. Also Hallauer, Bulletin de la Soc. Indust. de Mulhouse, vols. 47 to 53, 1877–1883. Also Peabody, chap. 17, Thermodynamics of the Steam Engine. Also Dwelshauvers-Déry, Revue Universelle des Mines, vol. 8, 1880, p. 362.

247. For Reeve's Temperature-entropy Chart, see Sidney A. Reeve: Entropy-temperature Analysis of Steam Engine Efficiencies, Progressive Age Pub. Co., New York, 1897.

250. For compressed-air engines, see Pernolet, Air Comprimé Röntgen-Dubois.

266. For hot-air engines, see Rankine, Steam Engine, § 270 to 278. Röntgen-Dubois; Ewing; also John Bourne, Examples of Steam Gas and

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280–303. For Gas and Oil Engine, consult Clerk, Donkin, Robinson, Hiscox. See also School of Mines Quarterly, January 1897: Eng'g News, April 8, 1897, p. 218.

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305. For Vapor Engines, consult

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315. For Wellington Series Engine, consult Engineering News, April 8, 1897 ; p. 210 : also p. 217, 394.

317. For Combined Vapor engines consult Rankine, Steam Engine, § 300.

325 *et seq.* On Refrigeration, consult

Ice-making Machines, by M. Ledoux, translated by J. E. Denton, D. S. Jacobus and Adam Riesenberger, 1897.

Trans. A. S. M. E. as follows :

Denton, Performance of 35-ton Absorption Machine, vol. 10, p. 792.
" " of 75-ton Compression Machine, vol. 12, p. 326.

Wood, Mech. and Phys. Properties of SO_2, vol. 12, p. 137.
" Some Properties of Ammonia, vol. 10, p. 627, vol. 11, p. 133.
" Test of a Refrig. Plant, vol. 11, p. 830.

Richmond, Note on Refrig. Process, vol. 14, p. 183.

Linde, Refrig. Machine of To-day, vol. 14, p. 1414.

340. For Injector, see

Strickland L. Kneass, Theory and Practice of the Injector, also Trans. A. S. M. E. as follows :

Webb, J. B., vol. 10, pp. 339, 888.

Peabody, C. H., vol. 11, p. 187.

See also Röntgen-Dubois, chap. 22, p. 531 : also Wood, Thermodynamics, p. 279 ; also Peabody, Thermodynamics, chap. 10, p. 145.

344. Flow of Steam in Pipes. A formula which has had some acceptance was published by Babcock and Wilcox in their catalogue " Steam." If W be the weight of steam in pounds, with a weight per cubic foot, D at an initial pressure p, then the loss of pressure from the resistance of a pipe of diameter d and of length L will result in a final pressure p_2 at the farther end of the pipe. The loss of pressure can be assumed and the diameter calculated, or the diameter assumed and the loss calculated. The formula is :

$$W = 300 \sqrt{\frac{D(p_1 - p_2)d^5}{L\left(1 + \frac{3.6}{d}\right)}}$$

The following table calculated by this formula for a length $L = 240\ d$ and a loss of pressure of one pound (or $p_1 - p_2 = 1$) is quoted from the same source:

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe in Inches. Length of each = 240 diameters.						
	¾	1	1½	2	2½	3	4
	Weight of Steam per minute in pounds, with one pound loss of pressure.						
1	1.16	2.07	5.7	10.27	15.45	25.38	46.85
10	1.44	2.57	7.1	12.72	19.15	31.45	58.05
20	1.70	3.02	8.3	14.94	22.49	36.94	68.20
30	1.91	3.40	9.4	16.84	25.35	41.63	76.84
40	2.10	3.74	10.3	18.61	27.87	45.77	84.49
50	2.27	4.04	11.2	20.01	30.13	49.48	91.34
60	2.43	4.32	11.9	21.38	32.19	52.87	97.60
70	2.57	4.58	12.6	22.65	34.10	56.00	103.37
80	2.71	4.82	13.3	23.82	35.87	58.91	108.74
90	2.83	5.04	13.9	24.92	37.52	61.62	113.74
100	2.95	5.25	14.5	25.96	39.07	64.18	118.47
120	3.16	5.63	15.5	27.85	41.93	68.87	127.12
150	3.45	6.14	17.0	30.37	45.72	75.09	138.61

Initial Pressure by Gauge. Pounds per Square Inch.	Diameter of Pipe in Inches. Length of each = 240 diameters.						
	5	6	8	10	12	15	18
	Weight of Steam per minute in pounds, with one pound loss of pressure.						
1	77.3	115.9	211.4	341.1	502.4	804	1177
10	95.8	143.6	262.0	422.7	622.5	996	1458
20	112.6	168.7	307.8	496.5	731.3	1170	1713
30	126.9	190.1	346.8	559.5	824.1	1318	1930
40	139.5	209.0	381.3	615.3	906.0	1450	2122
50	150.8	226.0	412.2	665.0	979.5	1567	2294
60	161.1	241.5	440.5	710.6	1046.7	1675	2451
70	170.7	255.8	466.5	752.7	1108.5	1774	2596
80	179.5	269.0	490.7	791.7	1166.1	1866	2731
90	187.8	281.4	513.3	828.1	1219.8	1951	2856
100	195.6	293.1	534.6	862.6	1270.1	2032	2975
120	209.9	314.5	573.7	925.6	1363.3	2181	3193
150	228.8	343.0	625.5	1009.2	1486.5	2378	3481

For any other loss than one pound pressure, the tabular value should be multiplied by the square root of that loss; for any other length than 240 diameters, divide 240 by the desired length expressed in diameters, and multiply the tabular or calculated figure by the square root of this quotient.

Recent investigations by Ledoux (Annales des Mines, vol. 2, 1892) present a formula for the friction coefficient f in the form

$$f = \frac{\left(T_0{}^{1.94} - P_1{}^{1.94}\right)\delta^5}{0.000535 \ L M_0 (M_0 - 0.000436 \ jdL)}$$

in which p is the pressure in atmospheres p_0 at the beginning and p_1 at the end ; δ is the diameter in meters and L the length in meters ; M_0 weight in kilograms supplied per second ; and j is the weight of condensation per square meter per hour. The mean value from a series of experiments for f was 0.0011. The Babcock experiments agree quite closely with the foregoing.

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